HVAC OPERATION AND AIR DISTRIBUTION STRATEGIES FOR THERMAL COMFORT AND ENERGY CONSERVATION IN MOSQUES IN HOT CLIMATE

BY

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DEDICATION

This thesis work is dedicated to my teacher Shyakh Mohammed Khawaja Al-Amin As-Shareef, my ever loving parents and family members

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"In the name of Allah, Most Compassionate, Ever-Merciful"

Praise be to Allah, Lord of the Worlds. To Him we belong and to Him shall we return. May He send countless blessings and peace upon the noblest of His creatures, His chief emissary, our leader, Prophet Muhammad, and upon all of his brethren of prophets and messengers, his family, and his companions.

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TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

LIST OF ABBREVIATIONS

MRT	:	Mean Radiant Temperature
PMV	:	Predictive Mean Vote
PPD	:	Predictive Percentage Dissatisfied
T_{sk}	:	Average Skin Temperature
M	:	Metabolic Rate
W	:	Work done
T_{cl}	:	Average Surface Temperature of Clothed Body
f_{cl}	:	Ratio of Clothed Surface Area
R_{cl}	:	Effective Thermal Resistance (R-value) of clothing
T_a	:	Air Temperature
h_c	:	Convection Heat Transfer Coefficient
T_r	:	Mean Radiant Temperature
h_r	:	Radiative Heat Transfer Coefficient
W_a	:	Air Humidity Ratio
W_{sk}	:	Saturated Humidity Ratio at the Skin Temperature

ABSTRACT

Full Name : Syed Samiuddin

Thesis Title : HVAC Operation and Air Distribution Strategies for Thermal Comfort and Energy Conservation in Mosques in hot climate

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In hot and hot humid climates as prevailing in Saudi Arabia, achieving thermal comfort in buildings requires the use of HVAC system which normally consume substantial amount of energy. In intermittent buildings like mosques which have unique functional characteristics, thermal comfort is usually achieved by continuously operating the HVAC system which may result in unnecessary use of energy if improperly designed and operated. Achieving thermal comfort becomes more challenging in mosques particularly when intermittent HVAC operation is considered together with the impact of air distribution systems on the thermal environment.

This research work was focused on investigating the impact of different air distribution schemes on thermal comfort of the occupants when using different HVAC operation strategies in a medium sized mosque by CFD (computational fluid dynamics) technique. A medium size mosque was modelled with appropriate data from literature and validated using the real-time energy consumption data. Then for thermal comfort analysis, three basic air distribution schemes were created with 7 models (3-CBAD, 2-TWAD and 2-UFAD) and appropriate CFD boundary conditions were included for meaning full results. For continuous operation of HVAC system, mosque building consumed 181 kWh/m^2 of annual energy. EnergyPlus results for thermal comfort showed PMV value of 0.4 which is acceptable. However, assessment of thermal comfort using CFD method revealed that only CBAD with M3 scheme achieved thermal comfort and in TWAD and UFAD the mosque building was over cooled severely. These results indicated the potential for energy conservation and thus intermittent HVAC operation was applied. This saved 30% of the total annual energy consumption by reducing the consumption from 181 kWh/m^2 to 127 kWh/m^2 consequently saving 35% of the total cooling energy consumption. EnergyPlus results for thermal comfort indicated that acceptable thermal comfort conditions were not achieved. However, assessment of thermal comfort using CFD method indicated that acceptable thermal comfort conditions were achieved in 6 of 7 air distribution schemes with M5 being most efficient. As a sensitivity analysis 5 (1-CBAD, 3-TWAD and 1-UFAD) more models were created and all of these models achieved comfort. Appropriate recommendations were made based on the results.

ملخص الرسالة

الاسم الكامل: سيد سميع الدين

عنوان الرسالة: استراتيجيات تشغيل أنظمة التكييف والتهوية في المساجد في المناطق الحارة الرطبة لتحقيق الارتياح الحراري وتوفير الطاقة

التخصص: هندسة معمارية

تاريخ الدرجة العلمية: ديسمبر 2014

إن تحقيق الارتياح الحراري داخل المباني في المناطق ذات المناخ الحار والرطب كما هو الحال في بعض المناطق في المملكة العربية السعودية، يتطلب الاستعانة بأجهزة التكييف والتي قد تستهلك طاقه كهربائية كبيرة في حال عدم تصميمها وتشغيلها بالشكل الأمثل. وتعد المساجد من المباني الفريدة من حيث متطلباتها وطرق تشغيلها حيث يحتاج المصلين للشعور بالسكينة والإطمئنان واالذي يتأثر بشكل مباشر بمستوى الارتياح الحراري، وللحصول على الأجواء الحرارية المناسبة مع توفير الطاقة في المساجد فانه يجب اعتبار استراتيجيات تشغيل متقطعه كبديل للتشغيل الدائم لأنظمة التكييف بالإضافة إلى الاعتبارات التصميمية لأنظمة توزيع الهواء.

يركزهذا البحث على دراسة الآثار المترتبة على طرق توزيع الهواء المختلفة على الارتياح الحراري واستهلاك الطاقة عند تطبيق استراتيجيات مختلفة لتشغيل نظام التكييف والتهوية في المساجد ذات الحجم المتوسط باستخدام برنامج (DesignBuilder) والذي يشمل برنامج حساب استهلاك الطاقة وبرنامج حساب حركة الموائع (CFD) بالإضافة إلى برنامج تقييم الارتياح الحراري . وقد تم نمذجة ومحاكاة لمسجد متوسط المساحة والتحقق منه باستخدام معلومات الاستهلاك الفعلي لطاقة مسجد متوسط المساحة ثم بعد ذلك تم تحليل الارتياح الحراري ، باستخدام ثلاث انظمة رئيسة لتوزيع الهواء من خلال سبعة نماذج أظهرت النتائج أنه عندما يتم تشغيل نظام التكييف والتهوية بشكل مستمر في المسجد فان الاستهلاك يصل الى $181 kWh/m^2$ سنوياً مع تحقيق جيد للارتياح الحراري بقيمة 0.4 =PMV بينما أظهر تقييم الارتياح الحراري باستخدام طريقة حركة الموائع المحوسبة تبايناً بين النماذج من حيث الارتياح الحراري . كما أظهر تطبيق التشغيل المتقطع لنظام التكييف والتهوية توفيراً في الطاقة بلغ 30% من اجمالي الطاقة المستهلكة في حالة التشغيل المستمر لأنظمة التكييف من خلال تخفيضها من $181 kWh/m^2$ الى $127 kWh/m^2$ بينما بلغ توفير الطاقة المستهلكة في التبريد حوالي 35% . و قد أظهر تقييم البيئية الحرارية باستخدام تقنية حركة الموائع المحوسبة (CFD) مستوى مقبولاً من الارتياح الحراري . وللحصول على تحليل أدق تم عمل نماذج بديلة للوصول إلى المستوى المطلوب من الارتياح الحراري ومن ثم صياغة توصيات مناسبة بناءاً على هذه النتائج.

CHAPTER 1

INTRODUCTION

1.1 Background

The present day world with growing global economy has seen an ever growing demand for energy in every sector be it residential, commercial, industrial, or transport. According to International Energy Agency (IEA), World total final energy consumption from 1971 to 2011 has increased significantly from 4,674 Mtoe in 1971 to 8,918 Mtoe in 2011 with maximum consumption from fossil fuels like oil, coal, natural gas etc. Figure 1.1 shows the world total final energy consumption of different sources of energy from the year 1971 to 2011[1]. This increase in demand has raised some serious concerns about the future availability of depleting non-renewable resources as well as the environmental impact including pollution and global warming due to the increase in greenhouse gases.

Electrical energy although a very expensive form of energy to produce; has seen an increase of more than 90% from 1971 to 2011. In 1971, 9.4% (of 4,674 Mtoe) of the world total final energy consumption was of electricity which increased to 17.7% (of 8,918 Mtoe) by the end of 2011(Figure 1.2). If electricity consumption by sector is assessed, building sector has seen the maximum increase with 44.1% (of 439 Mtoe) of the world total electrical energy consumption in 1971 to 55.8% (of 1582 Mtoe) in 2011 as per International Energy Agency's Key World Energy Statistics 2013 (Figure 1.3). In

2011, Electricity generation by the type of fuel, 41.3% was from coal/peat, 21.9% from natural gas, 4.8% from oil and the rest from nuclear, hydro and other renewable sources of the 22,126 TWh total world electricity generation making fossil fuels the largest source for electricity generation [1].

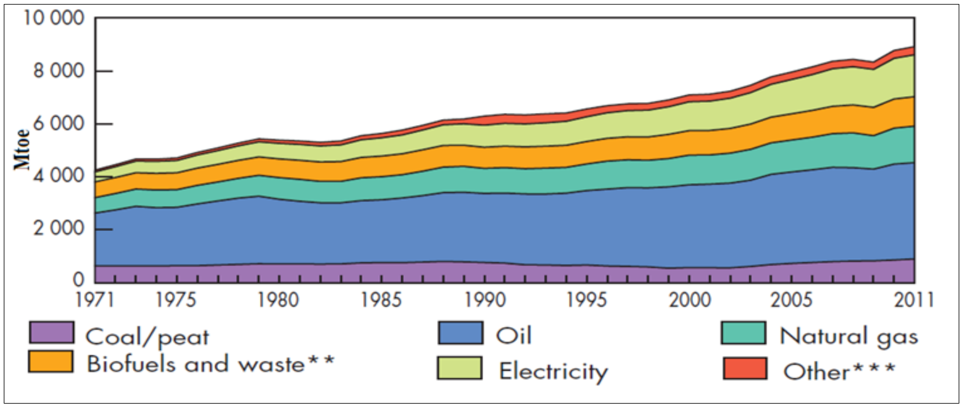


Figure 1.1: World total energy consumption from different sources from the year 1971 to 2011[1]

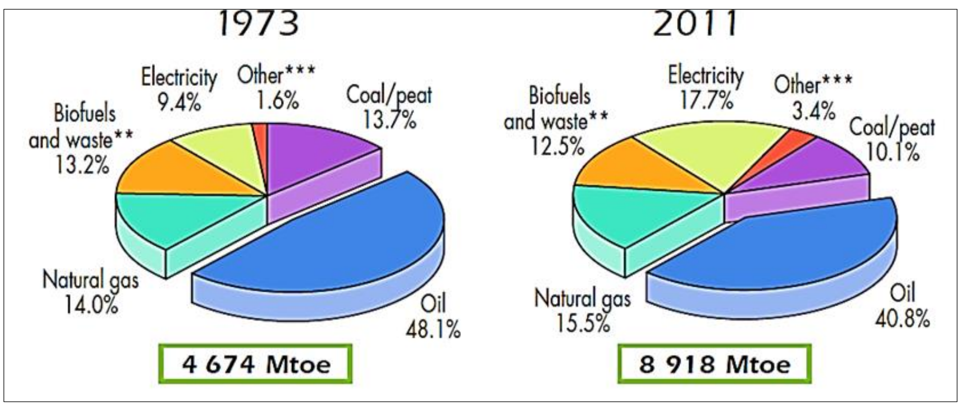


Figure 1.2: World total energy consumption from different sources by year 1971 and 2011 in %[1]

Focusing by the region, the Middle East has also seen a significant increase in energy consumption from about 0.7% (of 4,674 Mtoe) of the world total energy consumption in 1971 to 4.8% (of 8,918 Mtoe) in the year 2011. The primary source of energy for countries in the Middle East region in the present day is from fossil fuels like oil and natural gas. Middle East is the largest producer of crude oil of about 32.5% in the world by 2012 with Kingdom of Saudi Arabia producing 13.1% of the world's crude oil production, making it the largest crude oil producer in terms of countries. Furthermore Middle East is the third largest producer of natural gas in the world with 15.8% in 2012[1].

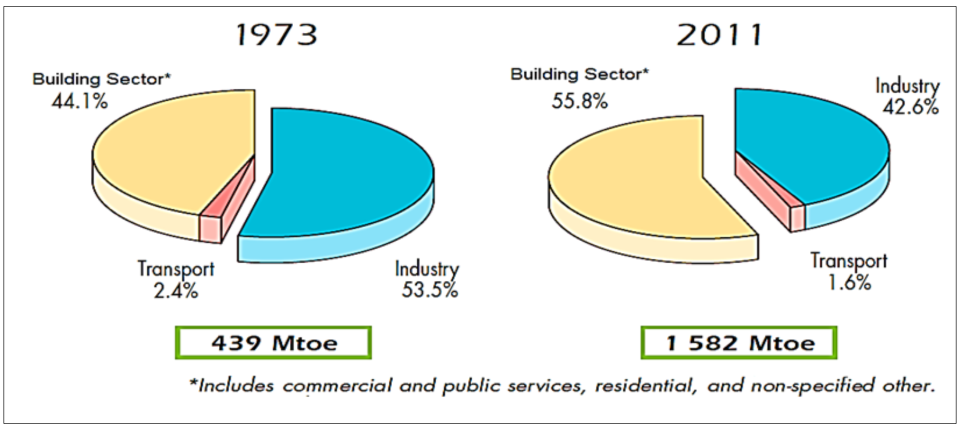


Figure 1.3: World total electrical energy consumption by sector in %[1]

In electricity generation, 3.8% (of 22,126 TWh) of the total world electricity generation was in the Middle East in 2011. And for its electricity generation, Middle East has relied primarily on crude oil and natural gas which are available in large reserves. U.S. Energy Information Administration's International Energy Outlook 2013 projections to 2040 has reported that world electrical energy generation will increase nearly to 40,000 TWh from 20,200 TWh in 2010 (Figure 1.4). And in Middle East it will increase to 1,405 TWh from 758 TWh in 2010 at growth rate of 2.1 percent per year on average in the reference case with similar fuel consumption trends were projected (Figure 1.5) reflecting region's rapid growth in population, economic activity, income and life style. Natural gas-fired electricity generation will rise at a 2.5-percent average annual rate while slowly displacing oil-fired electricity generation over the projection period. Oils' share of the region's electrical energy generation market will fall from 34 percent in 2010 to 14 percent in 2040 in accordance with the U.S. Energy Information Administration's International Energy Outlook 2013 projections (Figure 1.1)[2].

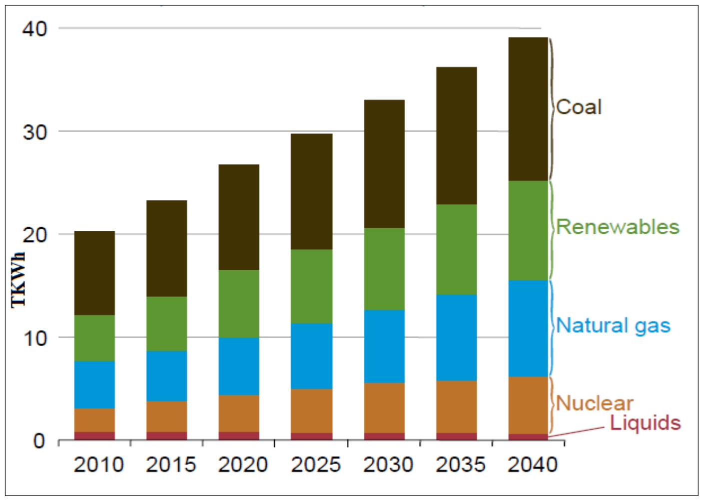


Figure 1.4: World net electricity generation by fuel, 2010-2040 (TKWh)[1]

According to Saudi Electric Company, electricity generation for the Kingdom of Saudi Arabia was 220 TWh in 2011[3]. International Energy Agency's Key World Energy Statistics 2013 reported that Kingdom of Saudi Arabia used oil to produce 142 TWh of electricity generation, second largest after Japan in 2011 making oil as a primary fuel source for Electricity generation. Electric Energy consumption in Kingdom of Saudi Arabia is growing faster than its GDP, leading to an increase in the total energy intensity (1.8 percent/year, on average, between 2000 and 2011) [3].

The electrical energy end use in Kingdom of Saudi Arabia is largest in building sector at 80% (residential, commercial and governmental). It is due to a fact that in the 21st century, the thermal comfort inside the built environment is achieved using HVAC system which requires electricity for its function implying buildings a major consumer of electrical energy (mainly for HVAC). Especially in hot climatic condition such as that prevailing in Kingdom of Saudi Arabia, where energy consumption is more dominant because of high cooling requirements using mechanical systems with HVAC system consuming 70% of the building electricity consumption implying that air conditioning percentage of Saudi electricity consumption is 52%[3]. There by increasing the carbon foot prints of the building by increased greenhouse gases emissions. This upward trend reflects that development in the country is based on energy-intensive industries, as well as on electricity-intensive lifestyles in buildings, encouraged by low electricity prices[3]. These consumption data and future projection for electric energy consumption show an ever increasing demand for electrical energy, especially in the Building sector, thereby developing an interest for concerned researchers to carry out their research in energy efficient building using energy conservation opportunities mostly focused to reduce the HVAC systems energy consumption. Thus the past decade has seen the world move towards more energy-efficient building designs, so has Kingdom of Saudi Arabia, especially to reduce HVAC system energy consumption, as an increase in energy efficiency of buildings contributes towards a significant decrease in greenhouse gases emissions and extends the life of available non-renewable energy resources.

Buildings like Mosques which represent a place of great stature where Muslims worship. They are considered unique buildings characterized by five intermittent operating schedules depending on the prayer timings and mode of prayer. Based on the mode of prayer the mosques are categorized in two basic categories: 1) “Daily” prayer mosques and 2) “Friday” + “Daily” prayer mosques. The five prayer timings vary throughout the year in accordance with the sun path, unique for every region in the world, thereby creating a distinctive operating schedule in a mosque. In order to achieve their intended function and provide a sense of peace and tranquility among worshipers, the indoor environment needs to be maintained comfortable during all prayers. In hot climates such as that prevailing in Saudi Arabia, thermal comfort in mosques is maintained through the use of HVAC system with a variety of HVAC systems like centralized systems, packaged system, split system (wall mounted and/or floor mounted), window units etc. The bulk of energy consumption in mosques goes to space conditioning with a typical mosque consuming 90% for HVAC system and 10% by Lighting and other equipment (like sound system, water coolers etc.).

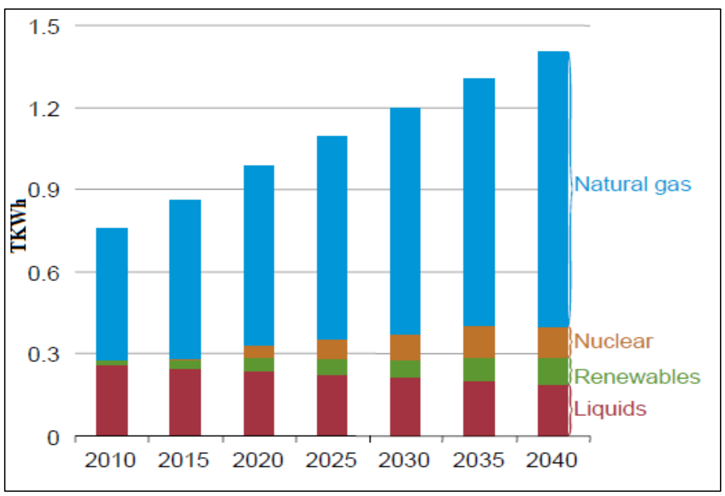


Figure 1.5: Middle East net electricity generation by fuel, 2010-2040 (TKWh)[1]

1.2 Statement of the Research Problem

Considering the unique operation and energy flow in mosques, it is evident that considerable amount of energy can be saved by proper air distribution design and operation of HVAC system without compromising the quality of the indoor environment. In different studies like that concluded by M.S. Al-Homoud et al.[4], I. Budaiwi et al. [5], Al-Ajmi [6] and Saeed [7] it was shown that substantial amount of energy will be saved in mosques when proper HVAC operation strategies are implemented. I. Budaiwi et al. [8] in their study of mosques which have intermittent occupancy schedule, found that energy savings can be achieved by employing intermittent HVAC operation provided a proper design and operation of the system considered. Thermal comfort might be compromised when implementing intermittent HVAC operation strategies by starting the HVAC system one hour before the start of each prayer and switching off when prayer ends unless a proper oversizing of HVAC system is employed [8]. They carried out energy simulation to support their argument and reported a savings of 23% in annual cooling energy by employing this one hour HVAC operation strategy along with proper system over-sizing. But not improvising on the fact that different air distribution strategies yield different thermal comfort patters and suggesting to oversize the system only based on thermal comfort results of the software simulation. It is a known fact that most of the simulation software considers cooling the whole volume of the building instead of just focusing on the occupied zone. Whether the thermal comfort is achieved by this operation strategy during the occupancy period without oversizing the system with necessary increase in operation time if required, in contrast with air distribution strategies is a subject of concern; owing to the fact that Mosques are large rectangular spaces characterized by high ceiling heights. Many studies have elaborated that when utilizing different HVAC air distribution strategies that can enhance the thermal comfort, is not necessarily be associated with high energy consumption. Ali Alajmi et al. [9]compared conventional ceiling-based air distribution (CBAD) with the under-floor air distribution system (UFAD) which uses different air flow patterns for an office building in hot climate. Their results suggest that thermal comfort is not altered but enhanced by different air-flow patterns and also energy consumption is reduced by significant amount by using UFAD. Gon Kim et al. [10] simulated the thermal environment of a large space with a high ceiling height using 3-D Computational fluid dynamics (CFD) software to investigate the effectiveness of UFAD in providing acceptable thermal environment for the occupants and its practical application to building. They used various supply air diffuser discharge velocities and locations of diffusers at a particular supply air temperature and reported enhancement in thermal environment. By implementing proper air distribution strategies, thermal comfort conditions can be improved when employing specific HVAC operation strategies. It is evident that in quest for energy saving while maintaining thermal comfort in buildings it is important to investigate the simultaneous impact of HVAC operation and air distribution strategy on energy consumption and thermal comfort in mosques.

1.3 Significance of the Research

Mosques are significant part of daily life in Kingdom of Saudi Arabia, the birth place of Islam. Due to their unique intermittent occupancy schedule and functionality Mosque buildings in Kingdom of Saudi Arabia requires a quality of thermal comfort standards for worshipers that is maintained using HVAC system. However, HVAC system is continuously operated often causing overcooling while consuming large amounts of energy then required in space conditioning and entirely depend on the availability of electricity. Most importantly, the primary/basic source for producing electricity in the Kingdom is oil which is a non-renewable energy source and out of the total energy available, about two-thirds of the energy is lost in producing electricity and one third remain in the form of electrical energy. Thus, there is a need to conserve energy in Mosque buildings that can be achieved by proper adjustment in the HVAC operation but there is a concern that thermal comfort might be compromised. A proper air distribution strategy has the potential to address this concern while using a proper operation strategy. Hence, studying Mosques buildings for the energy performance without compromising on the thermal comfort is necessary for their functionality and energy efficiency.

1.4 Objectives of study

The main objective of this research work is to:

- Investigate the impact of different HVAC air distribution strategies on thermal comfort for a particular HVAC operation strategies aiming at reducing energy consumption in Mosques.

- Recommend a combined HVAC operation and air distribution strategies leading to an acceptable level of thermal comfort with reduced energy consumption for a typical mosque.

1.5 Scope and Limitations

A proper HVAC operation and air distribution strategy is a sustainable and socially responsible approach to energy efficient and functional building design. The scope of this research is to utilize and investigate the effect of different air distribution strategies on thermal comfort inside a mosque during different occupancy schedules while employing intermittent HVAC operation strategy for energy efficiency of Mosques in Kingdom of Saudi Arabia. Intermittent HVAC operation strategy is an approach that improves the energy performance and when used in conjunction with different air distribution strategies reduces the energy demands without compromising on thermal comfort of the building occupants. This methodology gives the feasibility to use different air distribution design ideas leading to improvement in thermal comfort. As a result, all possible HVAC air distribution design and potential intermittent HVAC operation strategy work in tandem with each other in one single design of innovation. The thesis research work is limited to hot and hot-humid climates as characterized by the weather in Dhahran, Kingdom of Saudi Arabia. The scope of the work is limited to medium size mosques used for Friday and daily prayers and also the software limitations in predicting energy consumption and thermal comfort, which is a steady state model.

1.6 Research Approach

In order to achieve the stated objective, a research approach involving several steps is presented in a flow chart in Figure 1.6 and in step as follows:

Phase-I. Literature Review:

I. Literature review is carried out to acquire comprehensive understanding of the issues related to energy conservation, thermal comfort level and HVAC system operation and air distribution strategies.

II. Studies carried out in the field of thermal and airflow models and simulations and their related design elements.

III. Collect information about the mosques' common design practices, types of HVAC systems used, operation strategies and electricity consumption.

IV. Identify State of the art integrated CFD, Thermal Comfort and Energy Simulation tool

Phase-II. Formulation of Base Model and Validation:

I. Selected a software tool from group of available energy simulation programs for modeling the Mosque building in hot and hot humid climate.

II. Modelled the base case model of Mosque building by inputting the building characteristics obtained from literature review.

III. Simulated the base case model, and studying the energy performance and Thermal comfort for the Mosque building.

IV. Verified the Base case model for energy pattern using the data of an existing mosque building.

Phase-III. Investigated Impact of HVAC Air Distribution Strategies:

I. Investigated the impact of HVAC air distribution strategies on thermal comfort status when employing continuous operation of HVAC system in base case model Mosques.

Phase-IV. Investigated Impact of HVAC Air Distribution and Operation Strategies:

I. Investigate the impact of HVAC air distribution strategies for thermal comfort when employing HVAC operation strategies that reduces energy consumption in Mosques.

Phase-V. Conclusions and recommendation:

I. Recommended a combined HVAC operation and air distribution strategies leading to an acceptable level of thermal comfort with reduced energy consumption for a typical mosque.

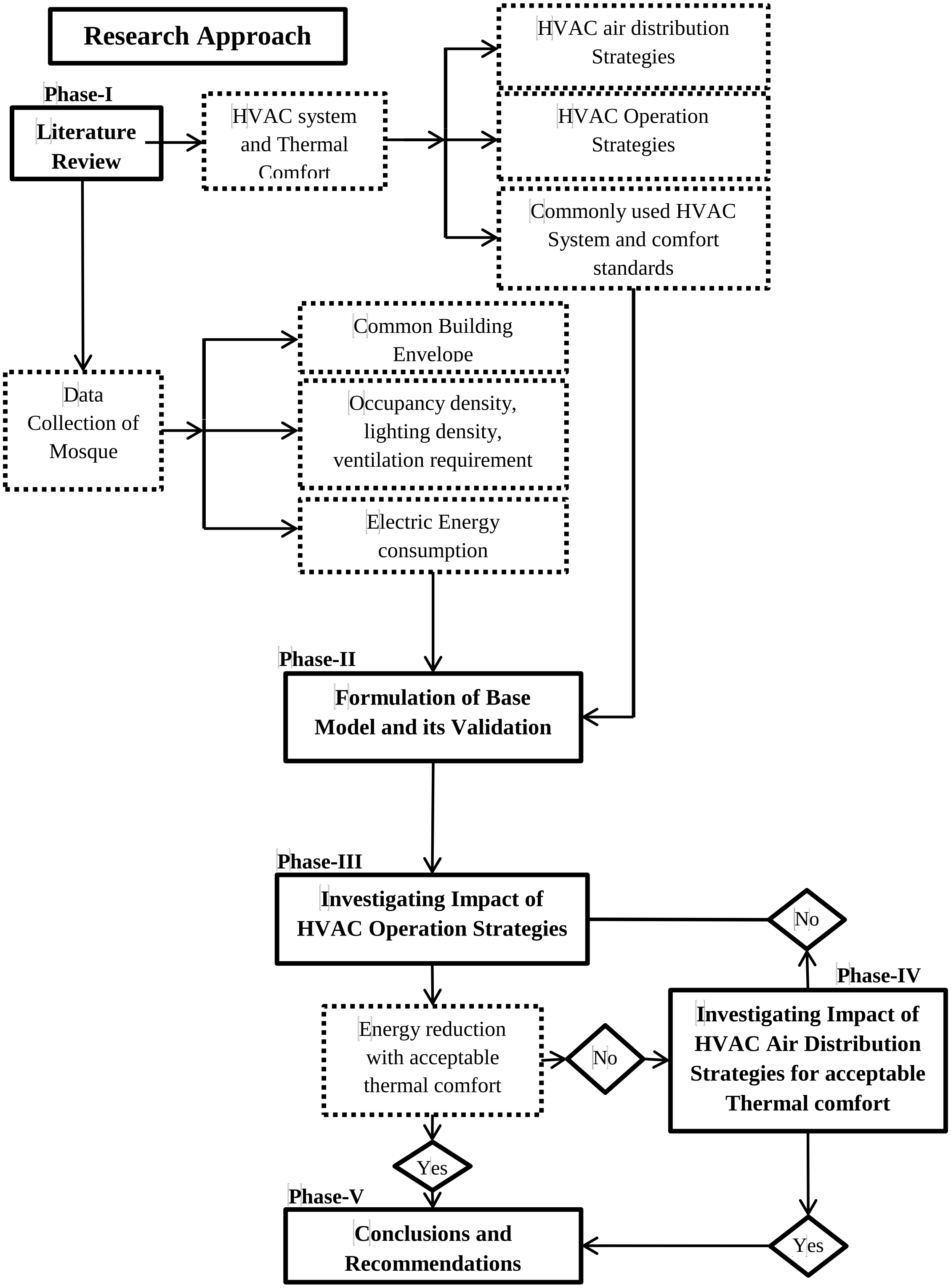


Figure 1.6: Research approach Flow chart

CHAPTER 2

LITERATURE REVIEW

The literature reviewed will provide an understanding of the Thermal Comfort criteria by highlighting the concepts of thermal comfort and its determining factor. And it will shed some light on the studies carried out by researchers on energy and thermal comfort issues in Mosque Buildings and their approach to solve these problems. Furthermore, the description and design elements of HVAC Operation and Air Distribution strategies will be addressed as the step towards achieving the above stated objectives.

2.1 Thermal comfort: Environmental factors and assessment tools

The main purpose behind the human creativity of built environment is to provide protection from the harsh outdoor climatic conditions. The built environment not only protects humans from the harsh outdoor climate but provide a certain degree of thermal comfort. In the 21st century, the thermal comfort inside the built environment is achieved utilizing HVAC system. There are verities of HVAC system available in the market today but on a basic level they are divided in to four types [11]:

1. **Window air conditioners**: Single rooms most commonly use this type of air conditioners. In this air conditioner one single box contain all the components, like the compressor, condenser, expansion valve or coil, evaporator and cooling coil. They have return and supply side by side and do not give flexibility for the use of different air distribution strategies. This air conditioner unit is provided in an opening made in the wall of the room, or more commonly a windowsill.

2. **Split air conditioners**: These consist of two parts: the outdoor unit which houses components like the compressor, condenser and expansion valve, fitted outside the room and the indoor unit that the evaporator or cooling coil and the cooling fan provided inside the room. Even these systems have return and supply side by side and do not give flexibility for the use of different air distribution strategies. This type of air conditioner can be used to cool one or two rooms and they can be either floor mounted or wall mounted.
3. **Packaged air conditioners**: This air conditioner is used when multiple rooms or a big space is needed to be conditioned. Packaged air conditioners consists the compressor, condenser, expansion valve and evaporator, packed in one box or sometimes the compressor and condenser are packed in one casing and expansion valve and cooling coil are placed in different casings which are located outdoor and indoor respectively. A high capacity blower fan is used to throw the cooled air which flows to different locations or rooms through the ducts layout giving flexibility for the use of different air distribution strategies like ceiling-based air distribution, underfloor air distribution or through the wall air distribution.
4. **Central air conditioners**: When buildings with multiple zones needs to be air conditioned, using individual units for each zone expensive forcing central air conditioners as a better option. Central air conditioning is used for cooling big buildings, big halls (like auditoriums, movie theaters, Mosques etc.), houses, offices, malls, huge spaces, galleries, entire hotels, gymnasiums, factories etc. The air distribution configurations like overhead ceiling air distribution, underfloor air distribution or through the wall air distribution are commonly used in these systems.

These varieties of HVAC systems provide different schemes of thermal comfort patterns by varying its determining factors. There are many studies in the literature that define and detailed the determining factors for predicting thermal comfort in a built environment. Many laboratory and field experiments were performed by the researchers to define conditions at which a specified percentage of occupants feel comfortable with the indoor thermal environment of a space. Estimating thermal comfort has a significant importance and organizations like ASHRAE (American Society of Heating, Refrigerating, and Air conditioning Engineers) provide the thermal comfort index for a built environment. ASHRAE in its Standard 55 Titled "Thermal Environmental Conditions for Human Occupancy" defined human thermal comfort in a built environment as ***"the state of mind which expresses satisfaction with the thermal environment"*** [12]. This standard establishes that thermal comfort consists of six determining factors categorized in two groups [12]:

1) **Personnel factors**: There are two factors in this group determining the characteristics of the occupants of the built environment. These factors often vary with type of the building and have a detailed definition for each type of building. These include:

    a) **Type of Activity or Metabolic Rate:** This parameter varies for every human with different type of built environments (residential, commercial, transport etc.). Definition of metabolic rate given by ASHRAE Standard 55-2010 is the amount at which the chemical energy is transformed into heat energy and mechanical work by metabolic activities within a human, frequently expressed as unit area of the total body surface. Unit used to express Metabolic rate is met units, which is converted as follows: 1met = 58.2 W/m^2 [4]. Popular values are 0.7 met for sleeping, 0.9 met for a seated and standing quiet positions, 1.1-1.4 met for light activities standing etc., [12, 13].

b) **Clothing Insulation or Amount of clothing:** The amount clothing worn by a person has a significant impact on thermal comfort, as it influences the heat loss from the body by acting as an insulator there by affecting the thermal balance. Generally, the insulating ability increases with the thickness of the garments. For winter wear the value is taken as 1clo and for summer wear as 0.5 clo where 1 clo is equivalent to 0.155 m^2 K/W [12-14].

2) **Environmental factors**: There are four factors in this group that determine the characteristics of the thermal environment. These are objective parameters that can be measured for the given environment. These include:

   a) **Air temperature:** It is the average dry bulb temperature of air surrounding an occupant, with respect to location and time. As per ASHRAE standard 55, ankle, waist and head levels are the spatial averages that are accounted, which vary for seated and standing occupants. Temperature stratification for air temperature at the head level being higher than at the ankle level may cause thermal discomfort. ASHRAE Standard 55 recommends that the difference not be greater than 3 °C. Since there are large variations from person to person in terms of physiological and psychological satisfaction, it is hard to find an optimal temperature for everyone in a given space. Comfort range for air temperature varies depending upon the air velocity inside the conditioned space; generally a variation between 21 ^oC to 26 ^oC is acceptable [12-14].

b) **Mean Radiant Temperature (MRT):** The radiant temperature is the amount of radiant heat transfer occurring between surfaces and occupants. The mean radiant temperature depends on the temperatures and emissivities of the surrounding surfaces as well as the view factor, or the amount of the surface that the object can see. Large differences in the thermal radiation of the surfaces surrounding a person may cause local discomfort or reduce acceptance of the thermal comfort conditions. ASHRAE Standard 55 sets limits on the allowable temperature differences between various surfaces as people are more sensitive to some asymmetries. The ceiling is not allowed to be more than 5 °C warmer than other surfaces, whereas a wall may be up to 23 °C warmer than the other surfaces [12-16].

c) **Air speed or velocity:** It is the average speed of the air to which the body is exposed, with respect to location and time. As per ASHRAE Standard 55, the comfortable velocity range is 0.10-0.25 m/s for uniform heat loss from the skin.

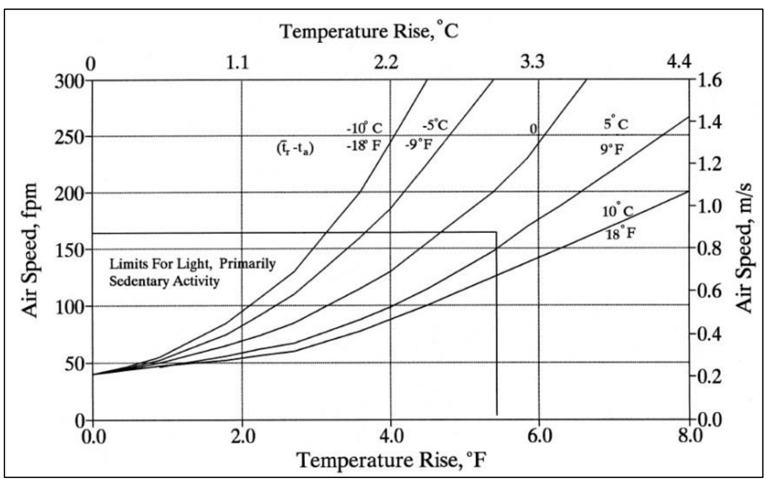


Figure 2.1: Allowable Air Temperature Rise

It is worth mentioning here that a particular relationship between increased air velocity and improvement in thermal comfort is yet to be established. However, ASHRAE standard 55 allows elevated air velocity which can be used to increase thermal comfort at higher air temperature values. Figure 2.1 shows the amount by which the air temperature can be increased to accommodate for elevated air velocities. The curves marked in the figure are the combinations of air velocity and temperature that results in equal heat loss from the skin and are presented for different values of a difference between MRT and air temperature. ASHRAE standard 55 allows a maximum of 0.8m/s air velocity in the occupied zone which can offset the air temperature by a maximum value of 3ºC. Clothing and activity level has a significant on the benefits that can be gained through air velocities. Higher activity level increase the skin wittedness, thereby increasing the effect of elevated air velocity compared to that in sedentary activity. And also the effect of increased air velocity is greater with increased amounts of exposed skin and lighter clothing [10, 12, 17].

d) **Relative humidity:** For a specific temperature and pressure, it is the ratio of the amount of water vapor in the air to the amount of water vapor that the air can hold. 30-60% is the recommended level of indoor relative humidity range in air conditioned buildings [12].

Based on these determining factors the thermal comfort in a conditioned space can be assessed using two main different models: the Fanger/static comfort model (PMV/PPD) and the adaptive comfort model.

2.1.1 Fanger/Static comfort model: PMV/PPD

The PMV/PPD model was developed by P. O. Fanger. It was published first in 1967 and then in 1972 and was the first one to be developed. To define comfort it uses empirical studies and heat balance equations concerning skin temperature. Fanger prepared a thermal comfort survey which asks subjects to evaluate their thermal sensation on a seven point scale from cold (-3) to hot (+3) (see Figure 2.2) [12-14, 17-19].

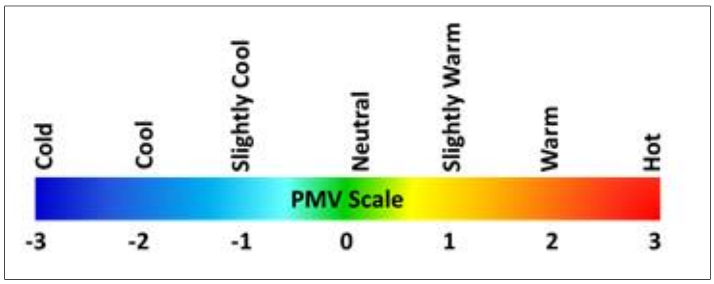


Figure 2.2: PMV scale[13]

Fanger provided equations that are used to determine the Predicted Mean Vote (PMV) of a large group of subjects at a particular combination of metabolic rate, clothing insulation, air temperature, mean radiant temperature, relative humidity and air velocity [12-14, 20]. These equations are based on the relationship between the temperature of skin and rate of sweating required for "optimal" comfort conditions. Total heat loss from the body is the difference between the metabolic generation and work done (e.g., walk, lifting) which can be expressed as:

$$q_{met,\ heat} = M - W \qquad \text{……………………….(1)}$$

However, when sedimentary or light activities of occupants are analyzed inside a built environment, the work done was found to be very small. After specifying these conditions, Fanger formed a correlation for PMV as a function of the thermal load on the body (**S**) which is the difference between metabolic heat generation rate and the estimated heat loss from the body to the actual environmental conditions that are assumed to "optimal" comfort conditions. The radiation and convection heat transfer will be functions of the clothing temperature, which depends on the temperature of skin. The skin temperature directly influences the evaporative losses from the body. Figure 2.3 presents a depiction of heat balance between human body and its surroundings. So the thermal load is as follows [21]:

$$\boldsymbol{S = M - W \pm R \pm C \pm K - E \pm Res} \quad \text{………………………(2)}$$

$$\begin{gathered} \boldsymbol{S} = q_{met.heat} - f_{cl}\, h_c\,(T_{cl} - T_a) - f_{cl}\, h_r\,(T_{cl} - T_r) - 156 \\ (W_{sk.\,req} - W_a) - 0.42(q_{met.heat} - 18.43) - 0.00077M \\ (93.2\, q_{met.heat} - T_a)\, q_{met.heat} - 2.78M\,(0.0365 - W_a) \end{gathered} \quad \text{………………………(3)}$$

$W_{sk,req}$ is the saturated humidity ratio which is calculated at the required skin temperature as the humidity ratio of air is in equilibrium with the skin at comfort conditions. In the above equations, the clothing temperature is not directly known. The temperature of clothing is not known directly and can be obtained from the required skin temperature, the thermal resistances, the air temperature and mean radiant temperature [21].

$$T_{cl} = \frac{T_{sk,req} + R_{cl} f_{cl}\,(h_c T_a + h_r T_r)}{1 + R_{cl} f_{cl}\,(h_c + h_r)} \quad \text{………………………(4)}$$

$$f_{cl} = \begin{cases} 1.0 + 0.2 I_{cl} & I_{cl} < 0.5clo \\ 1.05 + 0.1 I_{cl} & I_{cl} > 0.5clo \end{cases} \quad \text{………………………(5)}$$

$$T_{sk,req} = 96.3 - 0.156 q_{met,heat} \quad \text{………………………(6)}$$

$$h_c = max\begin{cases} 0.361(T_{cl} - T_a)^{0.25} \\ 0.151\sqrt{V} \end{cases} \quad \text{………………………(7)}$$

$$h_r = 0.7 \text{ Btu/h ft}^2\ {}^{o}\text{F} \quad \text{………………………(8)}$$

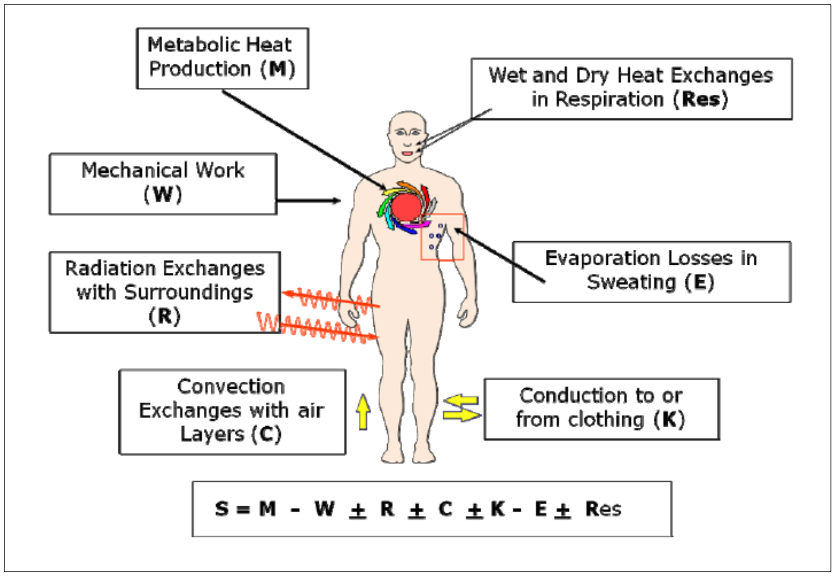


Figure 2.3: Human Thermal Comfort Heat Balance [13, 14, 21]

If observed closely, the clothing temperature is a function of **Operative Temperature** which defines the combined effects of radiative and convective heat transfer in an enclosure. It is defined as the weighted average of mean radiant and air temperatures by their respective heat transfer coefficients. As per ASHRAE standard 55 it can be approximated to a simply the average of mean radiant and air temperatures instead of a weighted average sedentary physical activities of the occupants (with metabolic rates between 0.9 met and 1.3 met) who are not exposed to direct sunlight and air velocities greater than 0.25 m/s. This relation advocates that the operative temperature is directly in proportion with the clothing temperature and variation in operative temperature will have a larger impact on the comfort conditions of the building occupants. At given values of air velocity, humidity ratio, clothing insulation and metabolic rate, a comfort zone has been defined by ASHRAE standard 55. This comfort zone uses a range of operative temperatures that can lead to an acceptable level of thermal environmental condition.

This combination has been used effectively for determining thermal comfort in the HVAC industry [12] and Figure 2.4show the same.

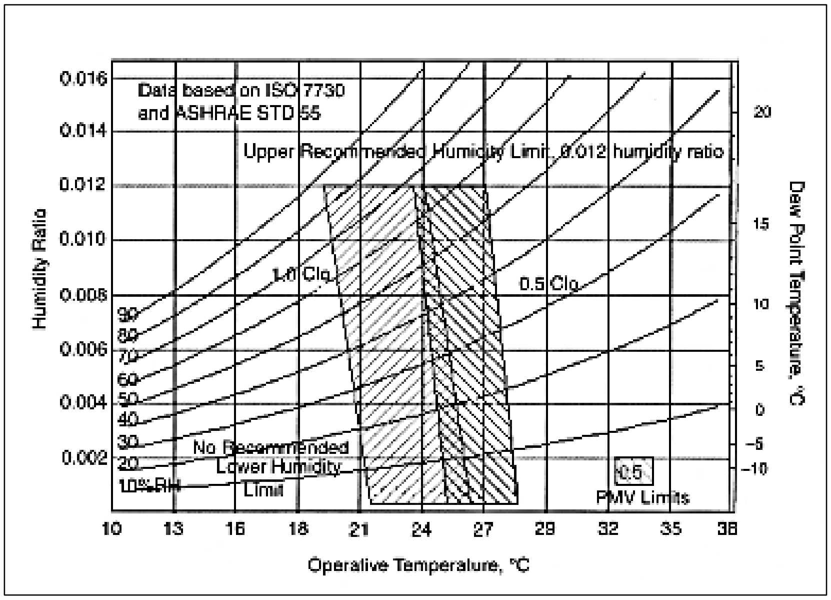


Figure 2.4: Acceptable range of operative temperature and humidity for spaces with 0.5 PMV [12]

Finally, Fanger developed the following correlation between PMV and the thermal load[13, 14]:

$$\textbf{PMV} = \mathbf{3.155\ (0.303}\boldsymbol{e^{-0.114M}} \mathbf{+ 0.028)\ S} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(9)$$

The fact that comfort has a subjective nature and would vary from person to person, thus predicted mean vote be the average response for a large number of people. Even though prediction of the thermal sensation of a large group of people is significant step in determining the conditions that are comfortable; however it is more useful to consider the satisfaction of the people. Thus Fanger developed another correlation in order to relate the PMV to the Predicted Percentage Dissatisfied (PPD) based on a study that surveyed several people in a chamber who's the indoor conditions were precisely controlled [12-14]. The empirical correlation between PMV and PPD with a graphical representation is as follows (see Figure 2.5):

$$\mathbf{PPD} = 100 - 95e^{-(0.03353PMV^4+0.2179PMV^2)} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(10)$$

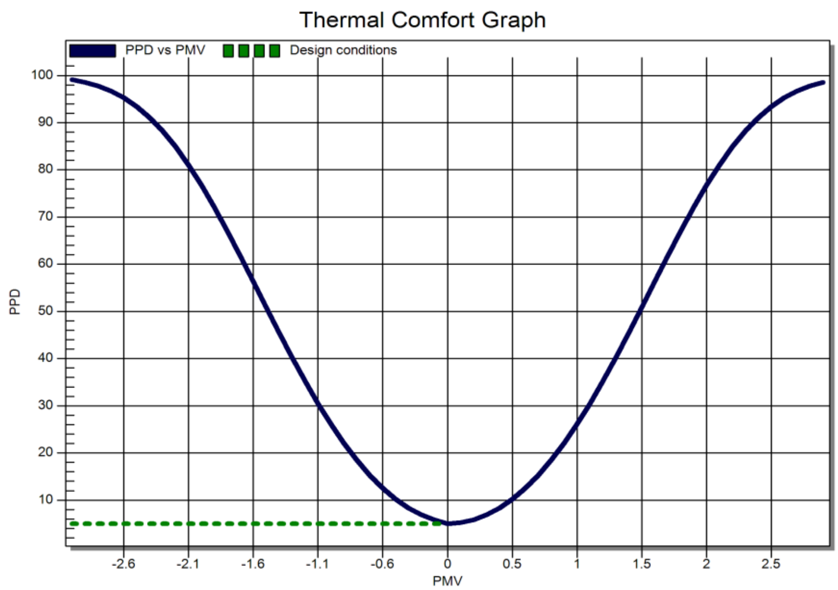


Figure 2.5: Thermal Comfort Graph with PMV and PPD

Ideal value would be zero that demonstrates thermal neutrality for the comfort conditions which is the combinations of the six parameters. Fanger model's commonly used PMV/PPD index values are:

I. -0.2 < PMV < +0.2 (PPD ≤ 6%)

II. -0.5 < PMV < +0.5 (PPD ≤ 10%)

III. -0.7 < PMV < +0.7 (PPD ≤ 15%)

IV. -1.0 < PMV < +1.0 (PPD ≤ 27%)

This method treats all occupants the same and disregards location and adaptation to the thermal environment [17, 19]. It basically states that the indoor temperature should not change as the seasons do. Rather, there should be one set temperature year-round. This is taking a more passive stand that humans do not have to adapt to different temperatures since it will always be constant. ASHRAE Standard 55 uses the PMV model to set the requirements for indoor thermal conditions and it requires that ***"at least 80% of the occupants be satisfied"*** [12].

There are other models available for thermal comfort prediction like the Pierce Two-Node model was developed at the John B. Pierce Foundation at Yale University which basically a modified version of the Fanger's PMV model. And the KSU Two-Node model, developed at Kansas State University which is based on changes in the skin wittedness that occur due to changes in the thermal conductance between the core and the skin temperature in cold environments, and in warm environments. But they are not very popular or preferred compared to the Fanger's PMV/PPD model [16, 19, 22, 23].

2.1.2 The Adaptive comfort model

The adaptive model is based on the idea that outdoor climate influences indoor comfort because humans can adapt to different temperatures during different times of the year. The adaptive hypothesis predicts that contextual factors, such as having access to environmental controls, and past thermal history influence building occupants' thermal expectations and preferences [24]. This model applies especially to occupant-controlled, natural conditioned spaces, where the outdoor climate can actually affect the indoor conditions and so the comfort zone. Numerous researchers have conducted field studies worldwide in which they survey building occupants about their thermal comfort while taking simultaneous environmental measurements. In fact, studies by de Dear and Brager showed analyzing a database of results from 160 of these buildings that occupants of naturally ventilated buildings accept and even prefer a wider range of temperatures than their counterparts in sealed, air conditioned buildings because their preferred temperature depends on outdoor conditions. These results were incorporated in the ASHRAE 55-2004 standard as the adaptive comfort model. The adaptive chart relates indoor comfort temperature to prevailing outdoor temperature and defines zones of 80% and 90% satisfaction [12, 16, 18, 19, 24-30].

2.2 Thermal comfort and Energy consumption in Mosques

Mosques represent a great place of importance and function for Muslims communities all over the world. Mosques are the central location where Muslims of all ages congregate for their daily as well as weekly “Friday” prayers. A feeling of tranquility and peace is what the worshippers are looking for in mosques but in order to attain that they need to feel thermally comfortable and relaxed inside the mosques. What is more unique about mosques is their occupancy schedule as compared to any other type of buildings. They are usually occupied five times intermittently during the day all year round but with different time for each prayer in different seasons or in other words, the prayer timing vary with the path of the sun. Each occupancy averaging a fraction of an hour to an hour where people gather gradually but not come to the mosque at a specific time during this hour and maximum occupancy occur during the actual performance of prayer which lasts for about 15–20 min and once the prayer is over, they gradually leave the mosque.

2.2.1 Energy Efficient Envelope Design

Generally mosques are built in a simple roofed rectangular shape walled enclosure prayer hall whose long side is normally directed towards Qibla (i.e. Ka'aba in the city Mecca, Kingdom of Saudi Arabia). Based on size and number of occupants, Mosques are divided in to three categories. 1) Small size mosques that can accommodate 40-250 people mostly used only for daily prayer, 2) Medium size that can accommodated 250-750 people, most of them used as both daily and Friday prayer and some as only for daily prayer and 3) Large size mosques that can house 750+ people and are mostly used for Friday prayer and rarely for daily prayer or may have operational zoning to use only a portion of the total area for daily prayer. They are usually characterized by high ceilings with a minimum value of 4 m to a maximum value of 12 m or above. Mosques-building styles and designs in Kingdom of Saudi Arabia, has been influenced and diversified by regional and cultural differences with a reflection of the environmental differences as well. The mosque architectural form, space, construction systems and building materials have evolved and developed to a greater extent particularly in Kingdom of Saudi Arabia. In addition, development in materials and environmental control systems (e.g. air-conditioning) have greatly influenced contemporary mosque architecture. Interior surfaces of contemporary mosques are mostly finished with reflecting materials such as plaster or marble, and the floor is usually carpeted. Hard painted concrete ceilings with a range of simple to elaborate decorations are commonly used. The window to wall ratio for mosques is around 15% as determined in the survey conducted by A. A. Abdou et al. for 132 mosques in eastern province of Kingdom of Saudi Arabia [5, 31, 32].

In Kingdom of Saudi Arabia, most mosques are equipped with air-conditioning system that consumes lot of energy because of the fact that most of them do not have thermal insulation in their building envelop as concluded by A. A. Abdou et al. in their survey results [31]. In spite of their pivotal role in the community and their unique characteristics, mosques received little attention in assessing their overall functional and operational performance. Several studies have investigated the acoustical performance of mosques utilizing measurements and simulation techniques. However, few studies have investigated the thermal and energy performance of mosques. As drive towards energy efficiency, mosques are now required to have thermal insulation in their roof and wall assemblies. Al-Homoud [33] conducted a simulation study for mosque envelope optimization and indicated that mosques design in the hot climates of Kingdom of Saudi Arabia should be air tight, well insulated, with light-colored surfaces and minimum area of shaded glass to avoid the dominant summer overheating. In another study Al-Homoud et al. [4]stated that acceptable thermal comfort conditions can be greatly enhanced by using thermal insulation in mosque envelope due to their façade dominated load, especially during the long un-occupied periods, as well as during their intermittent occupancy in their study to assess monitored energy use and thermal comfort conditions in mosques in hot-humid climates. In a study to address design retrofit in mosques I. Budaiwi et al. [34] recommended that wall and roof insulation should be used in all types of mosques especially future ones. They recommended a minimum thermal insulation value of 1.2 and 1.5 $m^2°C/W$ for walls and roof, respectively, should be used in mosques that could save up to 26% of the total energy consumption. And also stated; the mosque exterior finish should be treated to ensure low solar absorptance by using light colored surfaces [34]. In a similar study on mosques, I. Budaiwi concluded that as much as 25% reduction in cooling energy is achieved when using a moderate level of wall and roof insulation compared to an un-insulated envelope and recommended that a wall U-value of at least 0.7 $W/m^{2o}C$ (R = 1.4 $m^{2o}C/W$) and a roof U-value of about 0.5 $W/m^{2o}C$ (R = 2 $m^{2o}C/W$) is to be used in mosques[35].

2.2.2 Thermal Comfort in mosques

Mosques in Kingdom of Saudi Arabia rely on HVAC system of thermal comfort of its occupants in the very harsh hot climates that prevail in the kingdom. There are varieties of HVAC systems used in mosques with majority of the medium and large sized mosques using a central air conditioning system (mostly constant volume system) and small size and some medium sized mosques opting for a split or a window or a packaged system. The HVAC system in mosques is operated 24x7 in the hot climates across the Middle East so that thermal comfort can be maintained during occupancy. A limited number of studies have dealt with thermal comfort requirements in mosques. One such study on thermal comfort requirements for “Friday” prayer conducted in Riyadh by S.A.R. Saeed [7] during the hot summer season and reported that most of the worshippers indicated that they were comfortable, and only a few would have preferred cooler conditions. According to Al-Homoud et al. [4] when they investigated different mosques for energy use and thermal comfort, the relatively high energy use for some mosques is not necessarily translated into better thermal comfort conditions. During their monitoring of the mosques, thermal comfort is not achieved in most of the investigated mosques especially the un-insulated ones during times of peak thermal loads. F.F. Al-ajmi [6] conducted a study to investigate the indoor climate, thermal conditions and occupant thermal comfort sensations in six air-conditioned Mosque buildings evenly distributed across the State of Kuwait which is characterized by hot desert climate. The study surveyed a total of 140 occupants in six centrally air-conditioned Mosque buildings with ceiling air distribution system for human thermal comfort responses and also took the measurement of environmental parameters during the summer season of 2007. Conclusions of the study stated that the mean indoor dry bulb temperature of air was found to be 23ºC with standard deviation of 0.53 and mean relative humidity of 44.19% with standard deviation of 0.85 and a mean air movement of 0.23 m/s, with standard deviation of 0.08 in all Mosque buildings surveyed. 26.1 ºC was the neutral operative temperature during the Prayer which was obtained by linear regression analysis of actual mean vote on operative temperature. The Actual Mean Vote (AMV) was within the range -2.67 to 2.33 with the prayer's mean thermal sensation being 0.26, whilst for Predicted Mean Vote (PMV) the range was 0.005 to +1.43, with a mean PMV equal to +0.19. But when assessed individually mosque 1 had 23.87 mean air temperature with standard deviation of 0.20, mosque 2 had 20.27 mean air temperature with standard deviation of 0.35, mosque 3 had 21.53 mean air temperature with standard deviation of 0.3, mosque 4 had 19.13 mean air temperature with standard deviation of 0.95, mosque 5 had 28 mean air temperature with standard deviation of 0.93 and mosque 6 had 25.43 mean air temperature with standard deviation of 0.45. This shows that thermal comfort is not achieved in most cases, as most of the mosques were over cooled and significant reduction in the energy consumption of cooling energy without a corresponding loss of thermal comfort can be achieved by adaptation of an intermittent operation strategy for the HAVC system [6].

2.3 HVAC Operation strategies

The subject of thermal comfort in buildings is closely related to the issue of energy conservation. However, the desired thermal comfort may not be achieved due to the improper operation or control of the air-conditioning systems, resulting in under or overcooling of the space and possibly with a higher level of energy consumption than necessary. HVAC Operation plays a major role in the control of indoor environment and energy management. Many studies have been carried out on various building types to investigate this relationship and explore a means to conserve energy without compromising comfort. These have included investigations into the impact of various energy conservation measures, air-conditioning (A/C) systems and component characteristics on the thermal performance of building and thermal comfort of occupants. M. Fasiuddin et al. [36] similarly assessed the possible energy conservation opportunities from HVAC system operation strategies for commercial buildings in Saudi Arabia. Their study concluded that a saving of about 30% can be achieved for different combination of various HVAC operation strategies in commercial buildings while maintaining acceptable level of thermal comfort when implementing these strategies. Mohamad Fadzli Haniff et al. [37] conducted literature review of HVAC operation strategies for buildings towards energy-efficient and cost-effective operations. In their study, HVAC operation strategies were divided into three classes; 1) basic scheduling: this strategy operates the HVAC system by simply manipulating the ON and OFF states for the whole operating hours with a fixed set-point temperature. 2) Conventional scheduling: this strategy involves scheduling using pre-cooling or pre-heating techniques to reduce the peak demand with the use of several set-point temperatures throughout the operation cycle. 3) Advanced scheduling: it is the improved version of the basic and the conventional scheduling techniques and has proven that energy saving potential was higher than the conventional scheduling. They concluded that the demand reducing technique, which requires pre-cooling or pre-heating, is the most used operation strategy in HVAC scheduling among all operation strategies and expressed a concern over the exclusion of a human comfort index in implementation of these operation strategies. And recommended that in order to ensure a comfortable operation, some human thermal comfort indices such as the PMV or PPD are need to be considered in the new HVAC operation techniques.

2.3.1 Types of Operation Strategies

There are numerous available strategies applicable to the HVAC system of interest depending on the type facility. Below mentioned are a few strategies discusses by Manuel (1983) for energy management and control of indoor environment [38].

2.3.1.1 Time-Scheduled Operation or Intermittent Operation

This strategy consists of starting and stopping of the system based on the time and type of day. Type of the day refers to weekday, weekends and any other days that has a different schedule of operation. This is the simplest of all the energy conservation measure's function to maintain and operate. Another feature of time-scheduled operation of HVAC system serving areas not occupied 24 hours of the day is the optimized start/stop. HVAC system installed in buildings that has intermittent occupancy schedule should be shut down during un-occupancy and restarted prior to occupancy in order to cool down or heat up depending upon the requirements on a fixed schedule. This feature has the capability to automatically start and stop the system to minimize energy required to maintain the desired environmental conditions during occupied hours [38].

2.3.1.2 The Outside Air Temperature Cut-off

This strategy should stops the flow of cooling media upon the fall of outside air temperature to within 5° F of the inside design temperature. Similarly in event of provision of heating in the facility this function helps to cut-off the flow of heating media with the rise of outside air temperature to within 5° F of the inside design temperature [38].

2.3.1.3 Duty Cycling

This strategy involves shutdown of system for pre-determined short periods of times during normal operating hours. It is based on the principle that HVAC system seldom operates at peak output; thus if the system is shut off for a time, it has enough capacity to overcome the slight temperature drift that occurs during the shutdown. This function also helps to reduce the outside air cooling and heating load as the outside air damper is closed when the system is OFF [38].

2.3.1.4 Demand Limiting Start / Stop

This strategy helps in the reduction of electrical load that would add to setup peak electrical demand. There are numerous ways of accomplishing this task. Generally electrical loads are continuously monitored and predictions are made. When these predictions exceed the preset limits certain scheduled electrical loads are shut off to reduce the rate of consumption and predicted peak demand. The loads are turned off on priority basis, if the initial load drop action does not sufficiently reduce the peak demand [38].

2.3.1.5 Warm-Up / Night Cycle

The thermal load from the outside air used for ventilation contributes a substantial percentage of the total heating and cooling requirement for the building. This strategy helps to control the outside air damper when outside air is introduced during warm-up and cool down cycles prior to the occupancy and when the building is unoccupied [38].

2.3.1.6 Enthalpy Economizer

This strategy of using outside air economizer cycle can be a cost-effective energy conservation measure. This strategy utilizes the outside air to satisfy all or portion of the building cooling requirement when the enthalpy (total heat content) of the outside air is less than the return air from the space. The outside air is introduced into the building through the mechanical system during the cooling cycle in replacement for the recirculation air [38].

2.3.1.7 Space Temperature Night Setback

The energy required to maintain indoor space during un-occupancy, mostly for facilities not operating 24 hours/day can be reduced by raising or lowering the space temperature set-point, depending on the weather conditions [38].

2.3.1.8 Chilled Water Reset

The energy required generating chilled water in a reciprocating or centrifugal electric driven machines are influenced by number of parameters including the temperature of chilled water leaving the system. As chilled water temperature is selected for peak design times, in absence of effective humidity control, this temperature can be elevated during operating hours, in-order to satisfy the greatest cooling requirement [38].

2.3.1.9 Condenser Water Temperature Reset

Another parameter that affects the energy consumption by air-conditioning system is the temperature of condenser water entering the machine. In practice heat rejection system is designed to produce a specific condenser water temperature at peak wet bulb temperature. Optimizing of system can be attained by resetting the temperature to its initial value when the outdoor wet bulb temperature produces a lower condenser water temperature [38].

2.3.2 Application of Operation strategies to Mosques

Operation strategies because of their considerably high energy saving potential have become very popular among the energy conscious researchers in recent time. M.S. Al-Homoud et al. [4] studied indoor environmental parameters for mosques operated in the hot-humid climates of the eastern region of Saudi Arabia to determine the relation between thermal comfort conditions and energy use. Their study concluded that relatively high energy use in mosques can not necessarily translate into better thermal comfort conditions and suggested that proper HVAC operation strategy which is intermittent operation in combination with an appropriate operational zoning strategy will achieve significant energy savings with acceptable levels thermal comfort conditions. I. Budaiwi et al. [8] in their study of mosques which have intermittent occupancy schedule, found that thermal comfort might be compromised when implementing HVAC operation strategies that switches on the system one hour before occupancy for different prayer times and suggested for proper oversizing of HVAC system. They carried out energy simulation to support their argument and concluded that a savings up to 23% reduction in annual cooling energy is achieved by employing this HVAC operation strategy and system over-sizing. Comparing the cooling energy consumption of HVAC summer continuous operation of an un-insulated mosque with the consumption of the insulated mosque with properly oversized HVAC system operated for one hour before during each prayer, indicated that as much as 46% of cooling energy reduction can be achieved. They also provided a source design/operation guideline that has the information for professionals to improve the thermal and energy performance of mosques. But in their study they used only air temperature as a source for assessing thermal comfort.

These studies signify the role of HVAC operation strategies in the quest for energy conservation with a concern for human thermal comfort, especially when there are varieties of HVAC systems available in the market with each having their own air distribution strategies. So in order to justify the considerable amount of energy savings HVAC operation strategies can give, they need to be assessed in contrast with human thermal comfort with specific to HVAC air distribution strategies as it has an significant impact on environmental factors of human thermal comfort. Particularly in mosque buildings to employ Time-Scheduled Operation or Intermittent operation of the HVAC system, the effect of air distribution strategies should be known with this HVAC operation strategy in order to avoid over-cooling/heating or under-cooling/heating.

2.4 HVAC air distribution strategies

The type of system to use in different buildings have been unanimously defined by the HVAC engineers but the real confusion remain in which type of HVAC air distribution strategy to use for better thermal comfort with improved energy efficiency especially in the regions where use of natural ventilation is not applicable or over shadowed by mechanical ventilation. Very few studies are available in the literature that elaborate the impact of different air distribution strategies on the thermal comfort mostly focusing on the comparison between ceiling-based air distribution (CBAD) and underfloor air distribution system (UFAD) but none exploring the useful possibilities of through-wall air distribution system (TWAD). Ali Alajmi et al. [39] advocated that using UFAD system in commercial buildings significant energy can be saved. They used a custom made simulation tool specially designed for underfloor-air-distribution with EnergyPlus engine and concluded that UFAD system can save up to 30% energy compared to CBAD. They used PMV for occupant's thermal comfort assessment and stated that saving of energy in using UFAD was not prejudicing on occupant comfort. Jae Dong Chunga et al. [15] carried out a study for the thermal stratification which is crucial to system design, energy efficient operation and comfort performance of UFAD systems with an aim of examining impact of mean radiant temperature (MRT) on thermal comfort. They concluded that, keeping the same level of comfortable environment in the occupied zone, UFAD systems require much higher temperature of supply air, which represents significant energy savings, a clear elucidation of the benefit of UFAD systems has been shown by comparing it to the traditional overhead air distribution systems and stated that the benefit of UFAD systems is more pronounced at the condition of high ceiling height building. They found considerable discrepancies in thermal comfort for the assumption of using air temperature rather than MRT in of the evaluation of PMV and stressed that a more rigorous analysis is required to show any significant difference in PMV distribution for these two cases[15]. Qi Jie Kwong et al. [17] conducted a detailed review of thermal comfort assessment and potential for energy efficiency enhancement and concluded that the application of computational fluid dynamics (CFD) to study and predict the indoor environment was recognized to be useful and gaining popularity worldwide, although there were often some minor discrepancies with the actual environment that occurred because of the pre-determined boundary conditions. Besides, this computational method was noted to be a useful optimization and validation tool especially in studies related to energy savings improvement without compromising on thermal comfort, where the results of several studies presented the percentage of savings obtainable by comparison of different indoor scenarios. Thus, for building sector where physical measurements are difficult or less desirable, the CFD simulation may serve as a good alternative for field studies [17].

In the past decades, computational fluid dynamics (CFD) has been increasingly used as a prediction tool in the design and assessment of the indoor building environment. CFD is a group of numerical equations used to calculate fluid properties and predict probable air velocities, pressures and temperatures that will occur at any point throughout a predefined air volume in and around buildings with specified boundary conditions i.e. climate conditions, internal heat-gains, building constructions and window opening schedules etc. The equation set includes the three velocity component momentum equations (known as the Navier-Stokes equations), the temperature equation and where the k-e turbulence model is used, equations for turbulence kinetic energy and the dissipation rate of turbulence kinetic energy. Qiong Li et al. [40] used the CFD method to evaluate the indoor thermal environment of an air-conditioned train station building for three types of air-conditioning design schemes. They investigated impacts of air-conditioning design parameters such as supply air temperature, velocity, altitude and angle of incidence and their results indicated that analyzing the effects of air-conditioning design parameters on the building environment with CFD was an effective method to find the way to optimize the air-conditioning design scheme [40]. Son H. Ho et al. [41] used 2-Dimensional CFD tool to compare the thermal environment of two air distribution systems in an office setting. They modeled airflow, heat and mass (water vapor and contaminant gas) transfer in steady-state condition for an UFAD system with a CBAD and concluded that for both systems the PMV value does not change much due to different inlet location for UFAD system and inlet direction for CBAD system. The comparison of simulation results for the two systems shows that UFAD system has some advantages over CBAD system with an energy saving of 20-30% for UFAD system compared to CBAD which provides the same thermal comfort condition[41]. Ehab Mostafa et al. [42] used CFD to develop ventilation systems to prevent cold air drafts during the winter season and create a suitable atmosphere inside the broiler rearing building. In the cold weather, ventilation ducts and low ventilation rates are used to maintain the required air temperature. Four ventilation systems were designed in order to establish a comfortable zone for the broilers during winter season and by the use of CFD they improved the design with high uniformity ranging around 60-70% compared to the standard design. Wei-Hwa Chiang et al. [43] conducted a full-scale experiment in an office, and a computational fluid dynamics (CFD) simulation study for a radiant cooling ceiling system integrated with air dehumidification equipment installed in this test space. The obtained results from the experiment were compared with the values from the CFD simulation to validate the accuracy of the model. Predicted mean vote (PMV) index was used to assess the original indoor thermal condition and improved conditions according to the simulation results. Experimental variables included supply air temperature from the diffuser, surface temperature, and the area of the cooling panels. In addition, diffuser position in the mechanical ventilation system was analyzed that provided suggestions on improving the design of radiant cooling ceiling panels. This study proposed solutions alongside limitations on improving indoor thermal comfort and energy efficiency using a radiant cooling ceiling system in the subtropical regions of Taiwan. Computational technique is noted to be a useful method for optimization especially in studies related to energy savings improvement, where the results of several studies presented the percentage of savings obtainable by comparison of different indoor scenarios. Thus, for building regions where physical measurements are difficult or less desirable such as the regions that are subjected to building HVAC air distribution requirements, the CFD simulation may serve as a good alternative to field studies, especially for buildings like Mosques where High ceilings of mosques may cause stratification of heat above the occupied zone. Stratification is good for cooling and can be achieved by low elevation supply and return air where it does not mix with upper air. However, stratification is not good for heating and its effect can be reduced by using ceiling fans or low air distribution [12].

CHAPTER 3

BASE MODEL FORMULATION AND VERIFICATION

First this chapter will address the use of Building Performance Simulation (BPS) programs for whole building performance analysis to set familiarity with available tools and their characteristics. Highlights in this area include comparison of a number of BPS tools from the view point of their usefulness and applicability for HVAC Operation and Air Distribution strategies. Then this chapter presents the description of the mosque and focuses on the formulation of the base case simulation model. Mosques, in general, have simple geometrical configurations but vary in size. As discussed earlier that mosques are divided in to three basic categories based on their size and for the different types of mosques that are commonly built in Kingdom of Saudi Arabia, have a rectangular the geometrical configuration with similar thermal and operational characteristics. The mosque used in the study is a conceptual and energy efficient in its envelope characteristics with a depiction of the most common building design trends in the region. The chapter includes all information relevant to the developed base case model using the state-of-the-art software tool DesignBuilder as per the previous survey results conducted by researchers in the hot-humid region. It covers a wide range of data pertaining to building envelope systems, HVAC system, lighting system, energy consumption, assumptions if any, etc.

3.1 Building Performance Simulation (BPS) programs

To analyze energy performance of a building in design stage or for retrofits Building Performance Simulation tools are used increasingly as this tactic would help achieve a desired objective with different alternatives without compromising cost, energy and thermal comfort. The International Energy Conservation Code that aim to reduce the energy consumption of the building sector emphasize the use of Building Performance Simulation tools for both compliance paths i.e. perspective and performance paths to elaborate on how energy simulation can influence and inform the design process of buildings. In order to achieve the stated objectives, the first step is the determination of appropriate energy modelling software tool that had capabilities of energy simulation and an integrated CFD module with thermal comfort prediction. As advancements in software tools have aided in design decision making, one must never forget to consider the effect of modelling approach on buildings loads assessment and capabilities of the software tool. This otherwise will undoubtedly affect the decision making during the design phase. Conversion of real building geometry into an energy 38 model often results in neglecting and underestimating the translation effects. On the other hand, one of the main considerations to be given to the selection of a software tool is the analytical models or mathematical formulations on which its simulation engine is based. These define the capabilities of the software tool in question. Though there were many simulation engines earlier, it was DOE 2.1E that grabbed the attention and was widely used for a period of 30 years. Later U.S. Department of Energy started developing EnergyPlus that combined best features and capabilities of DOE 2.1E and BLAST [44]. Maile et al. [45] described the selection of energy simulation engine and discussed their usage over different life-cycle stages. The purpose of simulation engine is to support building design by comparing energy consumption of different design alternatives. Both DOE 2.1E and EnergyPlus provide such capabilities but differ from each other on various grounds [45]. Maile et al. [45] illustrated few functionality differences between them (see Table 3.1). Crawley et al. [46] contrasted the capabilities of building energy performance simulation programs with an up-to-date comparison of the features and capabilities of the most used building energy programs and was based on the following categories: general modelling features, zone loads, building envelope, 40 HVAC systems, electrical systems and equipment, economic evaluation, environmental emissions, etc. BLAST, BSim, DeST, DOE 2.1E, ECOTECT, Ener-Win, Energy Express, Energy-10, EnergyPlus, eQuest, ESP-r, IDA ICE, IES <VE>, HAP, HEED, PowerDomus, SUNREL, Tas, TRACE and TRNSYS were the building energy simulation programs that were considered [46].

Table 3.1: Functionality Differences between DOE 2.1E and EnergyPlus[45]

S. No.	Functionality	DOE 2.1E	EnergyPlus
1	Space load calculation method	Weight factor method	Heat balanced based approach
2	Loads & systems connectivity	No	Integrated loads & systems simulation
3	HVAC systems definitions	Predefined	Flexible; Component based
4	HVAC controls	Simplified representation	More flexible controls
5	New HVAC technologies	No detailed natural ventilation;	Moisture absorption & desorption; solar components; natural ventilation
6	Interconnectivity to other tools	None	Links to COMIS & SPARK
7	Time step	1 hour	Dynamic (ranges from 1 min to 1 hour)
8	Interoperability	No	Yes

Loads and systems connectivity functionality is very important as its availability in a simulation engine explains the inter-connectivity and integration of loads and systems at the time of simulation. This helps model real life scenarios which eventually lead to real time results. Weytjens et al. [47] compared six BPS tools namely ECOTECT, IES/VE – Sketch-Up, Energy10, eQuest, HEED and DesignBuilder based on the architect-friendliness. The focus of the study was energy performance of buildings to provide early design support for architects. Certain criteria were set to define the user-friendliness of the tools and the concluded that no tool was entirely adequate for architects use and it was observed that the simulation engine "EnergyPlus" is the developed trend in energy simulations and suggested that it must be used in building design. Worth noting here was the selection of DesignBuilder among the six tools for comparison. DesignBuilder provides a graphical user interface (GUI) to today's widely used energy simulation engine EnergyPlus [47]. Attia et al. [48] compared different BPS tools for architect-friendliness based on online survey which took into consideration ten tools, ECOTECT, HEED, Energy 10, DesignBuilder, eQUEST, DOE-2, Green Building Studio, IES VE, Energy Plus and Energy Plus- SketchUp Plugin (OpenStudio) and received 249 valid responses. Questions like one's position, tools they use, etc., were asked in the survey and highest numbers of responses were from architects and designers and many were from LEED accredited professionals. Two criteria's were set: 1) Usability and Information Management (UIM) and 2) Integration of Intelligent Design Knowledge-Base (IIKB) of the software tools. They concluded that architects and designers preferred IIKB over UIM in the tools interface with approximately 22% of the respondents saying they use DesignBuilder as a preferred tool for assessing energy performance of the buildings for both as a tool that was used in early design phase and during the retrofits. The tools were grouped into three categories and results revealed that DesignBuilder was ranked in the second category with a slightly less agreement among the respondents for architect-friendliness even though it was popularly known to have friendly GUI and varied graphical output features[49]. Attia et al. [49] compared ten early design tools, HEED, e-Quest, ENERGY-10, Vasari, Solar Shoebox, Open Studio Plug-in, IES-VEWare, DesignBuilder, ECOTECT and BEopt with the aim of using and integrating them during the design of NZEBs. Two criteria sets were considered; the first being a collection of five criteria namely usability, intelligence, interoperability, accuracy and design process integration, whereas the second being the design matrix for NZEB (see Table 3.2). DesignBuilder now in its latest version allows to model renewable technologies thus their comparison matrix is modified only for DesignBuilder.

These studies did not consider the possible use of CFD technique for whole building performance which in building design can be used to model the temperature distribution and movement of air within the spaces. It allows designers to investigate the temperature distribution and air movement within buildings before they are built, by allowing them to test different scenarios and implement the most effective solutions. CFD can be a useful tool for modelling:

- The thermal comfort of occupants.
- The distribution of environmental conditions within a space.
- The effectiveness of building services (such as the positioning of air inlets and extracts or radiators).
- The effectiveness of natural ventilation (such as the stack effect)

- The buildup of heat in specialist spaces such as server rooms.
- The positioning of sensors. For example in a tall space, the temperature at the top might be very different to the temperature at the bottom. This can be important when positioning temperature sensors that feed back to the building management system. Otherwise, heating and cooling might be operating unnecessarily.

CFD in itself only models air temperature and air velocity; but human thermal comfort within buildings is also dependent on radiant heat transfer occurring due to the different temperature of the surfaces inside the space. And when CFD is being used to predict human thermal comfort inside a space, it is necessary that both air temperature and mean radiant temperature are considered. A Building Performance Simulation tool with a CFD module which is able to include mean radiant temperature influences when calculating occupants' thermal comfort was considered for this study. From the above discussed software tools only DesignBuilder and IES VE-Ware has the CFD modules integrated with energy simulations. But in IES VR-Ware does not have a link between energy simulations and CFD simulation to consider real MRT values while it calculates PMV/PPD.

Table 3.2: NZEB Tools Matrix[49]

NZEB Criteria	**HEED**	**eQUEST**	**Energy 10**	**Vasari**	**Solar Shoebox**	**Openstudio**	**IES VE-Ware**	**ECOTECT**	**DesignBuilder**	**BeOpt**
Metrics	•	•	•	•	•	•	•	•	•	•
Energy	•	•	•	•	•	•	•	•	•	•
Environmental (CO_2)	•	•	•				•		•	•

Economic	•	•	•						•	•
Embodied Energy										
Urban Scale NZEBs										
Comfort & Climate	•	•	•		•		•	•	•	•
Climate Analysis	•	•	•	•			•	•	•	
Static	•	•	•	•			•	•	•	•
Adaptive					•					
Comfort Visualisation					•			•	•	
Passive Solar	•	•	•	•	•	•	•	•	•	•
Geometry & Massing				•	•	•	•			•
Daylighting	•	•	•				•		•	
Natural Ventilation	•		•				•		•	•
WWR		•	•				•		•	•
Thermal Mass	•		•				•		•	•
Shading Devices	•	•	•			•	•	•	•	•
Energy Efficiency	•	•	•	•	•	•	•	•	•	•
Envelope Insulation	•	•	•	•	•	•	•	•	•	•
Glazing Performance	•	•	•	•	•		•	•	•	•
Envelope Air Tightness	•	•	•				•	•	•	•
Artificial Lighting	•	•	•				•		•	•
Plug Loads	•	•	•				•		•	•
Infiltration Rate	•	•	•		•				•	•
Mechanical Ventilation	•		•						•	•
Cooling System	•	•	•	•			•		•	•
Heating System	•	•	•	•			•		•	•
Renewable Technologies	•		•		•		•			•
Photovoltaic (PV)	•		•		•		•		•	•
BIPV										
Solar Thermal			•				•		•	•
Innovative Solutions & Technologies					•		•			•
Mixed Mode Ventilation					•					
Advanced Fenestration							•		•	
Green Roofs							•		•	
Cool Roofs	•									
Double Skin Facades									•	
Solar Tubes										
Phase Change Materials									•	
CFD analysis							•		•	
MRT calculation									•	
PMV/PPD							•		•	

From the analysis of Table 3.2 and based on the objectives of this thesis work, DesignBuilder scores as a tool that is used in this study.

3.1.1 DesignBuilder

DesignBuilder is a user-friendly modelling environment where a user can work (and play) with virtual building models. It provides a GUI to today's widely used energy simulation engine EnergyPlus and its output features are popularly known to have varied graphical representations. It has strong design features that address the design aspects of environmental performance data such as: energy consumption, carbon emissions, comfort conditions, daylight illuminance, maximum summertime temperatures and HVAC component sizes etc., which holds good for carrying out parametric and performance based analyses. Maile et al. [45] illustrated the strengths, weaknesses and data exchange capabilities of DesignBuilder on four grounds namely tool architecture and functionality, life-cycle usage, data exchange and interoperability and limitations and concluded that the simulation program had most comprehensive user-interface for the most widely used energy simulation engine EnergyPlus. They portrayed the information workflow in a graphic form in DesignBuilder (see Figure 3.1). The workflow starts with selecting a location for carrying out the analysis. Then the tool allows the creation of building geometry and other definable parameters such as internal loads, construction types, windows, doors, lighting, material selection, HVAC systems, etc. As per DesignBuilder some typical uses of their software are [50]:

- Calculating building energy consumption.
- Evaluating façade options for overheating and visual appearance.
- Thermal simulation of naturally ventilated buildings.

- Reporting savings in electric lighting due to use of natural daylight.
- Prediction of natural daylight distribution through Radiance simulations
- Visualization of site layouts and solar shading.
- Calculating heating and cooling equipment sizes.
- Detailed design of HVAC and natural ventilation systems including the impact of supply air distribution on temperature and velocity distribution within a room using CFD
- ASHRAE 90.1 energy models
- UK, Ireland, France and Portugal Building regulations and certification reports
- Communication aid at design meetings.
- An educational tool for teaching building simulation to architectural and engineering students.

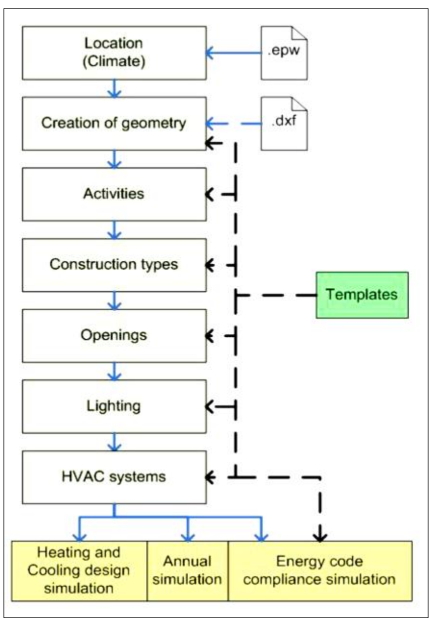


Figure 3.1: Information Workflow in DesignBuilder [45]

3.1.2 DesignBuilder CFD Module

DesignBuilder CFD module is its useful and excellent feature used for both external and internal analyses throughout a predefined air volume in and around building spaces with specified boundary conditions which may include the effects of climate, internal heat gains and HVAC systems. External analyses provide the distribution of air velocity and pressure around building structures due to wind effect and this information can be used to assess pedestrian comfort, determine local pressures for positioning HVAC intakes/exhausts and to calculate more accurate pressure coefficients for EnergyPlus calculated natural ventilation simulations. External analyses can only be conducted at the site level. Internal analyses provide the distribution of air velocity, pressure and temperature throughout the inside of building spaces and this information can be used to assess the effectiveness of various HVAC system designs and to evaluate interior comfort conditions. Internal CFD analyses can be conducted at zone, building block and building levels. Calculations can also be conducted for single zones that span several blocks by connecting them with holes and using the 'merge zones connected by holes' model option setting. One of the good feature of DesignBuilder CFD is if a model is created for the purpose of an energy analysis (e.g. thermal simulation, etc.), exactly the same model can be used for CFD analysis. The numerical method used by DesignBuilder CFD is known as a primitive variable finite-volume method, which involves the solution of a set of equations that describe the conservation of heat, mass and momentum. The equation set includes the three velocity component momentum equations (known as the Navier-Stokes equations), the temperature equation and where the k-e turbulence model is used, equations for turbulence kinetic energy and the dissipation rate of turbulence kinetic energy. The equations comprise a set of coupled non-linear second-order partial differential equations having the following general form, in which ϕ represents the dependent variables:

$$\frac{\partial}{\partial t}(\rho\phi) + \mathrm{div}(\rho u\phi) = \mathrm{div}(\Gamma\ \mathrm{grad}\ \phi) + S \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(11)$$

The $\frac{\partial}{\partial t}(\rho\phi)$ term represents the rate of change, the term div($\rho u\phi$) represents convection, the div(Γ grad ϕ) term represents diffusion and S is a source term. Due to the non-linearity, the equation set cannot be solved using analytical techniques, which necessitates the requirement for a numerical method. The numerical method in DesignBuilder consist of re-modelling the differential equations into a set of finite difference equations by dividing the calculation domain (or building space) into a set of non-overlapping adjoining rectilinear finite volume grid. These grids in DesignBuilder CFD are in Cartesian form and allow non-uniformity between major grid lines which are parallel with the major axes. Calculation process in DesignBuilder was developed to ensure convergence to the iterative solution of the set of equations if the equation coefficients were constant. However, the equation set is non-linear and the coefficients actually contain the dependent variables themselves, and consequently convergence cannot be guaranteed in all cases. The finite difference equation set is formulated in the form of a transient equation set although the calculations are steady state, i.e. essentially a "snap-shot" in time. It uses false time steps mechanism which is essentially the time step used in the pseudo-transient term for the equation of dependent variable to ensure that they change slowly. This method is known to be very effective relaxation method in order to arrive at a stable solution. Usually a 'best-guess' false time step is generated automatically for velocities in case of forced convection and for buoyancy driven flows DesignBuilder uses a default value of 0.2. Reducing the false time step has the effect of slowing down the change in the dependent variable and can be a helpful remedy for unstable solutions. However, DesignBuilder also allows Relaxation factor method to achieve convergence.

When it comes to modelling turbulence the following turbulence DesignBuilder uses two models namely **Constant effective viscosity** and **k-e** [50]. For this study **k-e** model was used as it is more accurate. CFD calculations require discretization of the Navier–Stokes equations which is a reformulation of the equations in such a way that they can be applied to computational fluid dynamics. Several methods of discretization are used and the **Upwind, Hybrid** and **Power-Law** discretization schemes are available in the DesignBuilder.

Each of the dependent variable equations requires meaningful values at the boundaries of the calculation domain in order for the calculations to generate meaningful results throughout the domain. Thus boundary conditions in DesignBuilder can import from the energy simulations that have been computed in a previous EnergyPlus simulation. The data that can be imported are inside surface temperatures of walls, roofs, floors, ceilings, windows, partitions, doors etc. This option gives the reliable results in CFD analysis as EnergyPlus provide close to reality inside surface temperatures. Currently boundary conditions data for internal CFD calculations can be imported on hourly bases.

Internal analysis boundary conditions tend to be more involved and can require the addition of zone surface boundaries such as supply diffusers, extract grilles, temperature and heat flux patches and also the incorporation of model assemblies representing occupants, radiators, fan-coil units, etc. The boundary condition types are available for surfaces of all orientations are **Supply diffuser, Extract grille/diffuser, Temperature patches and heat Flux patches.** And for ceiling and other downward-facing surfaces only additional boundary types for multi-directional diffusers like **Four-way** and **Two-way** supply diffusers are available. Additionally, CFD assembly library that is provided with DesignBuilder contains a number of pre-defined assemblies that can be used to add items such as occupants, radiators and furniture and some of these pre-defined assemblies already have CFD boundary attributes associated with them, e.g. occupant assemblies have a defined convective heat flux of 45W/m^2. Apart from this, thermal boundary type can be defined as **"None"** when a component block does not act as a CFD boundary condition, **"Temperature"** when a component block acts as a fixed temperature CFD boundary condition and **Flux** when a component block supplies a fixed heat flux to the surroundings. Figure 3.2 demonstrate CFD workflow in DesignBuilder.

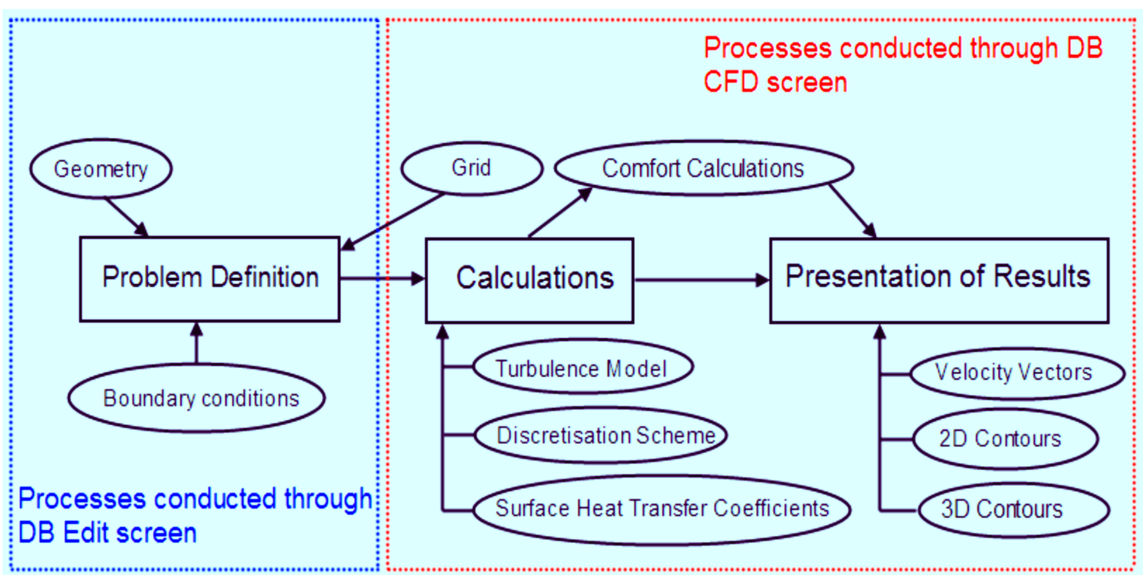


Figure 3.2: CFD workflow in DesignBuilder [50].

3.2 Mosque Building Characteristics

Table 3.3: Building Characteristics and Specifications

Characteristics / Specification	Description of the Mosque
Location	Dhahran (26.27 N latitude, 50.15 E longitude, and 17m above sea level)
Orientation	Oriented 245º from North
Shape	Rectangular
Floor to Ceiling Height	5.5 m
Floor Area	(24x20) 480m^2
WWR	15%
Infiltration	0.5 ACH
Occupancy	500 People
Occupant density	1.06 People/m^2
Metabolic rate	0.9 mat
Clothing Insulation	0.7 for Summer and 1 for winter

To accommodate a study that could be implemented to mosques that are used daily and for Friday prayer as well, a medium sized rectangular mosque with an area of 480 m^2 was modelled that have a capacity of 500 occupants. The rectangular shape of the mosque has an aspect ratio of approximately 1:1.2 with its long length at an angle of approximately 25º from the east-west axis. An occupant density of 1.06 People/m^2 was used that was obtained from relevant literature that had a detailed auditing of existing mosques done by researcher in the region. Table 3.3 describes the characteristics and specifications of the modelled mosque building.

3.2.1 Occupancy Characteristics

The occupancy is dependent on prayer times which vary throughout the year depending on solar time. In order to simulate variations in loads and scheduling time for the different thermal events, the year is segmented into four periods, each 3 months long, which roughly represent seasonal variations.

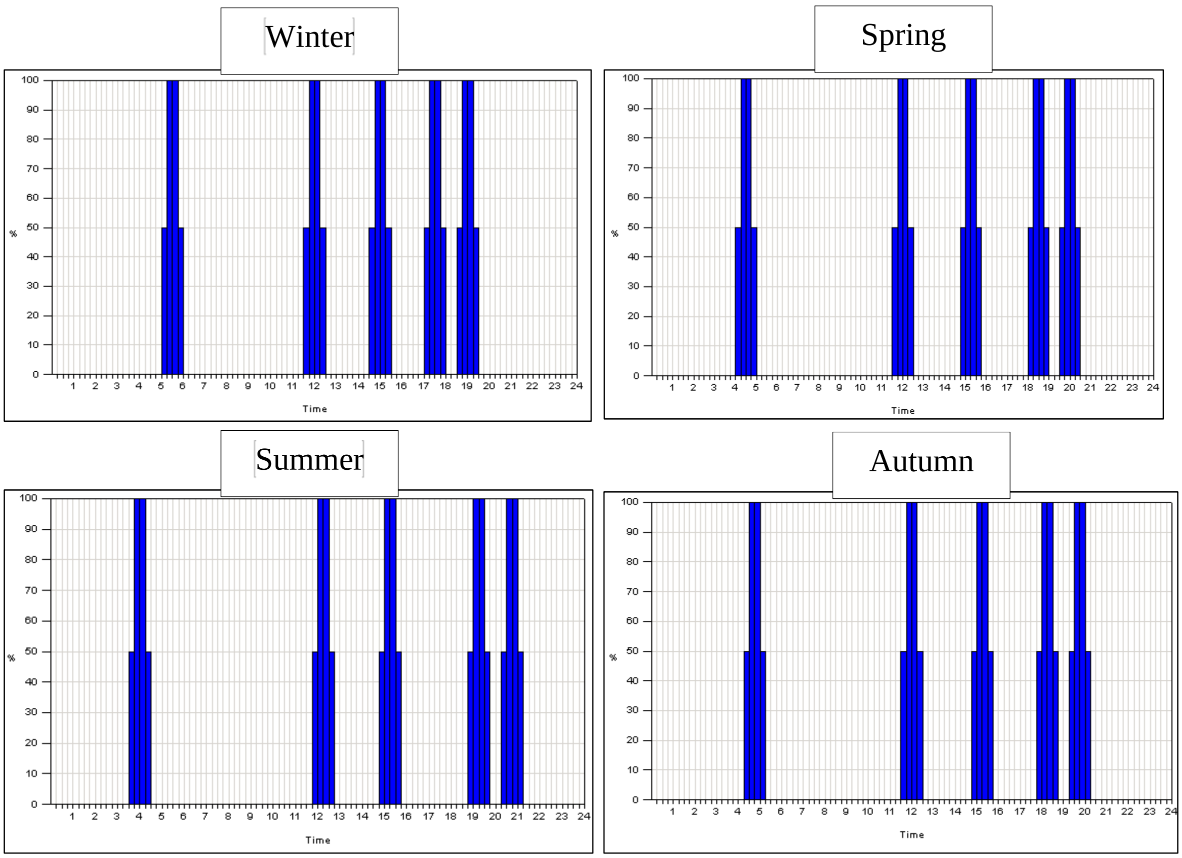


Figure 3.3: Occupancy Profiles for Different periods

The winter period includes the months of December, January and February; the spring period includes the months from March till the end of May. The summer period represents the summer period starting in June and ending in August and finally, the autumn period is from September until the end of November. The occupancy profile for each of these periods is shown in fig. 3.1. Occupancy is an important input parameter for a building's energy simulation. Mosques have a unique occupancy schedule as it randomly varies within the hour and differs from one prayer to another. It is, therefore, very difficult to accurately define an occupancy profile which characterizes the exact variations in occupancy load and patterns and considers the incremental increase in the number of occupants from the call to prayer (Azan) until the occupants leave the mosque. It is assumed that occupancy reaches maximum during each prayer in order simulate a worst case scenario that may occur during the month of Ramadan.

3.2.2 Building Envelope Information

The walls of the mosque have the following specifications: plaster (dense) as the outermost layer, concrete block (medium) on both side with thermal insulation sandwiched in between, and plaster (lightweight) as the innermost layer. The total thickness is 273 mm with an overall U-value of 0.656 W/m2-K. The concrete blocks have been observed to be equal in thickness; however, the thickness of the plaster is varying depending on its placement in the wall assembly. The roof of the mosque has the following specifications: roofing concrete tiles as the outermost layer, Cement Mortar, Sand/Screed Sloping, thermal insulation, Asphalt Roofing Roll, reinforced concrete (dense) and Plaster, lightweight as the innermost layer. The total thickness is 348 mm with an overall U-value of 0.466 W/m2-K. The windows of the mosque are of the sliding panel / fixed glass plate type in an aluminum frame without thermal break. They are double glazed with two glass layers sandwiching the air layer. Glasses are light tinted and the thickness of the two glass layers is different. The total thickness is 24 mm with an overall U-value of 2.709 W/m2-K. The flooring system of the mosque is a slab on grade. It has the following specifications: carpet fibrous pad as the outermost layer, glazed ceramic tiles, cement mortar, dense reinforced concrete, high density polyethylene, and sand as the innermost layer. The overall U-value is calculated to be 0.792 W/m2-K. Table 3.4 describes the assemblies accordingly.

Table 3.4: Building Envelop Characteristics

Envelope System Type	**Layers (Outside to Inside)**	**Thickness (m)**	**U-value (W/m^2-K)**
Wall	Plaster, dense	0.013	0.656
	Concrete Block, medium	0.10	
	Extruded Polystyrene	0.05	
	Concrete Block, medium	0.10	
	Plaster, lightweight	0.010	
Roof	Concrete Tiles, roofing	0.025	0.460
	Cement Mortar	0.013	
	Extruded Polystyrene	0.06	
	Asphalt Roofing Roll	0.025	
	Reinforced Concrete	0.2	
	Plaster, lightweight	0.025	
Window	Glass, generic tinted	0.006	2.709
	Air Gap	0.012	
	Glass, generic tinted	0.006	
Floor	Carpet	0.013	0.792
	Ceramic Tiles, glazed	0.013	
	Cement Mortar	0.013	
	Reinforced Concrete, cast-dense	0.125	
	Polyethylene, high density	0.002	
	Earth, gravel	0.5	

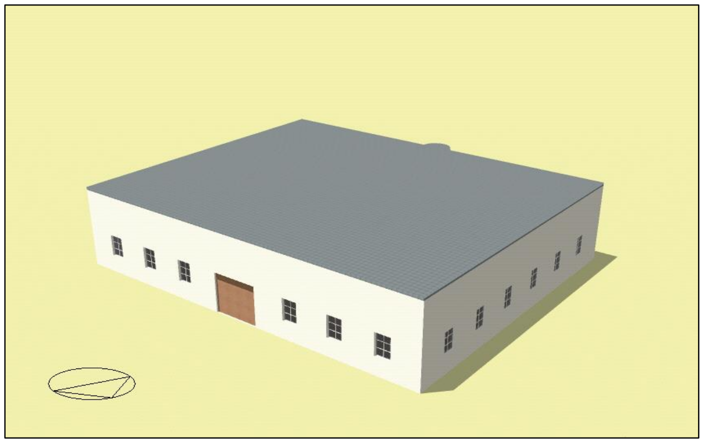


Figure 3.4: 3-D Rendered Image of Base Model

3.2.3 Building Cooling system

Table 3.5: Cooling system characteristics

Characteristics / Specification	Description of the system
HVAC System Type	Constant Volume system
Ventilation	2.5 L/s/Person + 0.3 L/s/m^2
Minimum supply Temperature	12^{o}C
COP	2.6
Operation	Continuous
Set-Point Temperature	24^{o}C

Mosques are a single zone building with a uniform cooling requirement unlike other buildings, so a constant volume system is selected for the building with a ventilation rate of 2.5 L/s/Person plus 0.3 L/s/m^2 as required by the ASHRAE Standard 62.1-2007: Ventilation for Acceptable Indoor Air Quality, in the category of religious places of worship. Minimum supply air temperature of the system is 12^{o}C with a COP of 2.6 and the set-point temperature of the space is 24^{o}C with continuous operation during all the months of cooling requirement.

3.2.4 Lighting System Information

Table 3.6: Base Case Lighting Information.

Characteristics / Specification	Description of the system
Lighting Power Density	12 W/m^2
Type of Lighting	Florescent Tube Lights
Operation	Fully on during whole Occupancy period

ASHRAE/IES 90.1 provides limits on the amount of lighting power installed in the building, expressed in watts per square meter, to promote efficient technology and design. The lighting power densities (LPDs) applied to interior of the mosque building is 12 W/m^2. Fluorescent tube lights were selected for fulfilling the required illumination level in the base case model, because they produce more light for a given amount of electricity and also they are long-lasting and have up to 10 times the lamp life of standard incandescent lamps. And also the common practice, florescent tube lights are used inside the mosques in the region and so is the case in this study. It is scheduled to operate fully during the complete occupancy period and remain close during non-occupancy.

3.3 Base Model Verification

Figure 3.4 shows the 3-D rendered image of the base case model. Base case model was created using all the data mentioned above and then simulated for energy consumption pattern. It was observed that August recorded the monthly highest and February recorded the monthly lowest shows that the envelope behavior was according to standards. However, before the base model can be used for the analysis; it needs to be verified for consistency of its results with an actual mosque building. The mosque building which is chosen for the comparison is Abu-Ubaidah mosquer located in the hot-humid climate of

Al-Khobar, in eastern region of Kingdom of Saudi Arabia, which is not far from the Dhahran, the location used for base model. The data for this building was obtained from a survey of mosques in Al-Khobar city by I. Budaiwi et al. in year 2002. The building is a rectangular shaped with its entrance facade facing 25^{o} from east direction. The dimensions of the building are 42.83 m length x 30.45m width x 5.m height. This building has a construction characteristics of a good thermally insulated mosque consists of single zone. The total floor area of the building, as obtained from building plans, is 1266 m^2. This has a capacity of 1306 occupants and the lighting power density is 12 W/m^2. Split system is used as an air conditioning system for this building with a COP of 1.8 and operated during prayers only. Table 3.7 illustrates the characteristics of the existing mosque building. The occupancy for this building is different from that assumed in the base case model. Table 3.8 shows the occupancy characteristics as surveyed [5].

Table 3.7: Characteristics of the existing mosque building[5]

Characteristics	**Description**
Location	Al-Khobar, Saudi Arabia
Type of building	Mosque
Plan Shape	Rectangular
Total height (m)	5.3
Floor Area (m2)	1266
Overall WWR (%)	15
Building orientation	245^{o} from NORTH
Occupancy Density (person/ m2)	1.06
LPD (W/m2)	12
HVAC	Split System
COP	1.8

Table 3.8: Occupancy characteristics of the existing mosque building[5]

Total Capacity of the Mosque	**% of Attendants During Different Prayers**					
	Fajr	Dhuhr	Asr	Maghrib	Isha	Friday
1306	10	10	20	25	30	100

Since the base case assumptions were for a worst case scenario, which is very much different from that of the monitored mosque, there were large difference between base case and surveyed mosque. Thus necessary adjustments were made to the base case model in occupancy schedule, infiltration rate, Operation and COP of HVAC system. The electric energy consumptions for the surveyed mosque were in total but not in segregated form. And the area of the two mosques differ significantly, therefore the comparison is made based on the electric energy consumption per unit area. Figure 3.5 illustrate the comparison of the two mosques. The measured values are for the year 2002 and the weather data used for the simulation of the base case model is of the year 2012. The large deviation in the months of March, April and November were due to the difference in weather condition for these months in the stated years. The monthly average temperature for March and April were higher in the year 2012 compared to that in 2002 and also monthly average temperature November was higher in 2002 compared to that in 2012 thus causing large variations. Monthly average temperatures for different months in stated years are presented in Table 3.9. The variations in other moths were minimal and within the allowed limits. An overall percentage deviation of 12% was observed between the electricity consumption of existing building and base case model which concludes that the model is reliable for evaluating the effects of energy conservation measures for the building under study.

Table 3.9: Monthly average Temperatures for year 2002 and 2012 [51].

	Jan	**Feb**	**March**	**April**	**May**	**June**	**July**	**Aug**	**Sept**	**Oct**	**Nov**	**Dec**
2002	15.7	16.9	19.7	25.7	34.0	36.4	38	36.7	33.2	29.5	23.3	18.6
2012	15.7	17.3	22.2	26.5	34.5	34.6	37	35.9	32.6	29.5	22	17.9

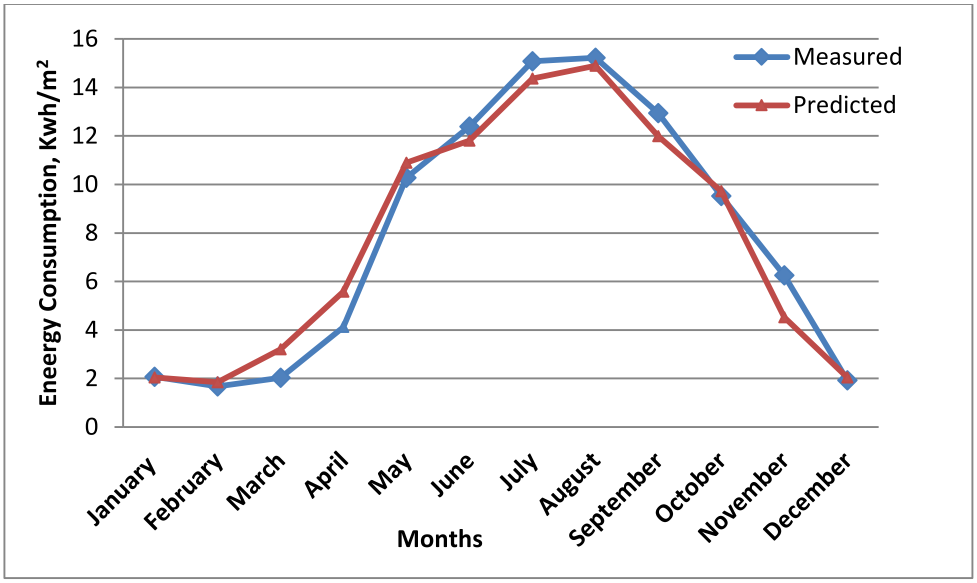


Figure 3.5: Electric energy consumptions comparison

CHAPTER 4

CFD MODELLING

4.1 Thermal Load Modelling

This part of the research work utilizes CFD module of state-of-the-art software DesignBuilder. The base case model is prepared for CFD simulation by including all the thermal loads that are available in the space including occupancy which is at 500 people inside the mosque building. Component assemblies that mimic a standing occupant are placed inside the space in a manner that is similar to actual occupancy during prayer. Figure 4.1 shows the component assembly of an occupants and Figure 4.2 represents the axonometric view of the whole space with occupant assemblies. This component assembly has a heat flux of 45W with a surface area of 1.55 m^2 and surface properties are different on each part to mimic clothing as shown in Figure 4.1.

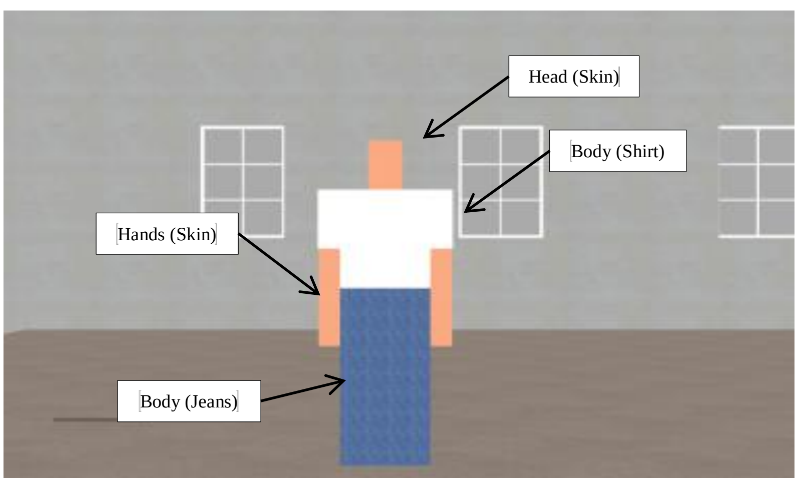


Figure 4.1: Occupant Assembly with surface properties

The lighting fixtures on the ceiling that add heat to the space which are included in the form of heat flux patches on the ceiling. From the energy simulation the total heat load due to lighting was found to be 5420W which is divided into 8 patches. The patches used for CFD are $1m^2$ with a flux 677.5W. Figure 4.3 shows the heat flux distribution that was used in CFD simulations.

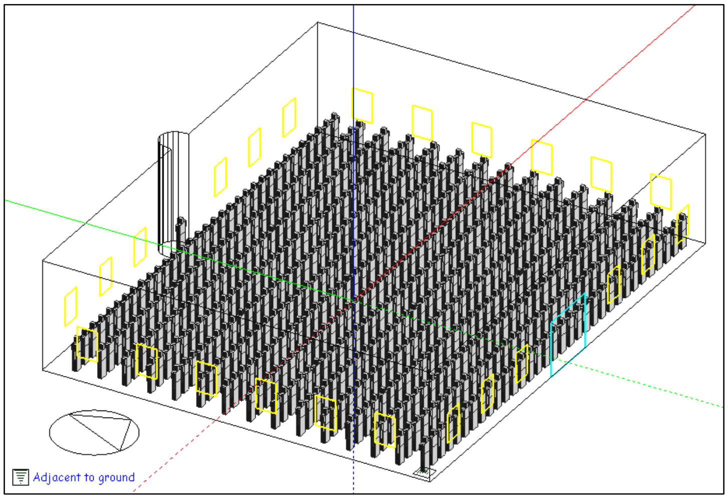


Figure 4.2: Axonometric view of Space with Occupant Assemblies

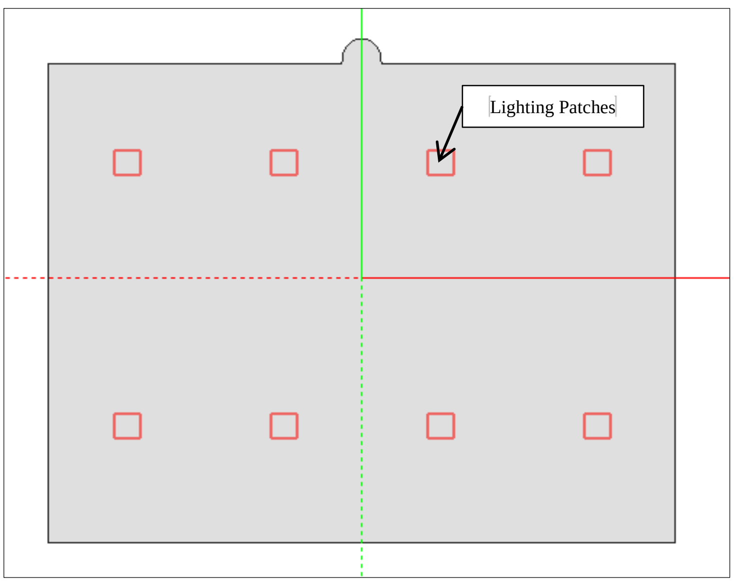


Figure 4.3: Lighting Heat Flux Patches

The boundary conditions which are the most essential for reasonable CFD simulation to be valid and need to be specified as close to reality as possible for accurate results. In the literature, it was observed that there were often some discrepancies in CFD simulation results that were presented in different articles with the actual environment that occurred because of the pre-determined temperature boundary conditions which are not realistic. These discrepancies can be minimized by obtaining more realistic temperature boundary conditions from energy simulation results of EnergyPlus in the form of inside surface temperatures.

DesignBuilder provides a link between EnergyPlus simulations and CFD boundary conditions that allows importing inside surface temperature of each surface at 1 hour time steps. For base case, EnergyPlus sized the HVAC system at the peak load which occurred on 21st July at 03 PM. And also the HVAC system selected is a constant volume system which increases the temperature of the supply air when there is a variation in the space load except at the time of peak load. Thus the HVAC system would work at its peak by supplying air at its lowest temperature of 12^{o} C at this time of the day. So the CFD boundary conditions are obtained for 21st July at 03 PM which corresponds to full occupancy time of Asr prayer for both continuous and intermittent HVAC operation cases. The volume flow rate for the HVAC system was found to be 5398 l/s or 5.398 m^3/s and the same was divided into equal value for each diffuser by dividing it by number of diffusers. In predicting thermal comfort a humidity ratio of 51.67% was used that was obtained from energy simulation for the base case at stated time and date for continuous HVAC operation and 36.15% was used for intermittent HVAC operation case.

4.2 Air Distribution Schemes Modeling

4.2.1 Ceiling-Based Air Distribution (CBAD) (M1, M2 and M3)

This is the most commonly used air distribution system especially for spaces such as mosques. There are two types of diffusers configurations with different layout in this system and each will be discussed individually.

4.2.1.1 CBAD with four way supply diffusers (M1)

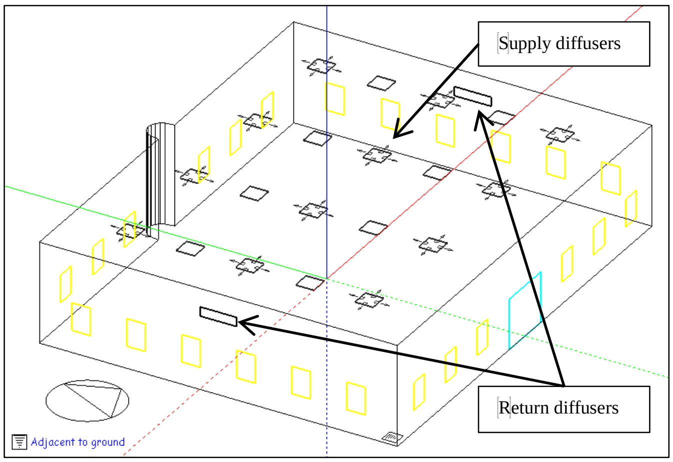


Figure 4.4: M1- Axonometric view of the M1 configuration

This configuration is most widely used in mosque building as the space is usually rectangular which can be divided into number of squares or rectangles of almost equal sides and a diffuser is placed at the center in each of these areas. Thus the same concept was used in this case and 12 diffusers were selected. The volume flow rate in each diffuser was 449.833 l/s. The return diffusers were provided on the two side walls with equal extract volume of 2699 l/s. Figure 4.4 shows the complete configuration and describes the location of return diffuser on the wall. The discharge velocities used are 1.5m/s, 2m/s, 3m/s and 3.5m/s because these velocities would fall well within noise criteria for most of the diffuser types.

4.2.1.2 CBAD with linear/slot supply diffusers and Ceiling Return (M2)

This configuration is again most commonly used in mosque building as the space is usually rectangular and diffusers are placed along the perimeter on the ceiling but at least at a distance of 1m from the nearest wall. Thus the same concept was used in this case and 14 linear/slot diffusers were selected. Volume flow rate in each diffuser was 385.57 l/s. The return diffuser was provided at the center of the ceiling, with same area as in M1 model which would represent a return diffuser type that is commonly provided in the lighting fixtures. Figure 4.5 shows the diffusers layout for this configuration and describes the location of return diffuser on the ceiling in axonometric view. The discharge velocities that were used are 1.5m/s, 2m/s, 3m/s and 3.5m/s same as previous case as theses velocities would fall well within noise criteria for most of the diffuser types.

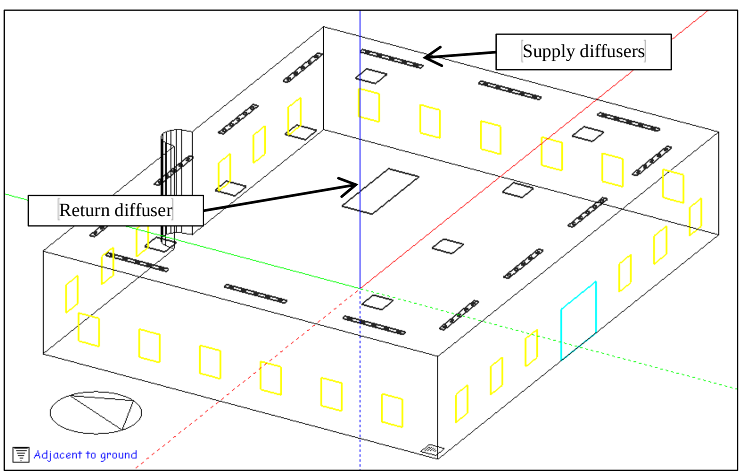


Figure 4.5: M2- Supply and return diffuser layout

4.2.1.3 CBAD with linear/slot supply diffusers and Wall Return (M3)

This configuration has found its practical application in buildings with high ceiling height with diffusers placed along the perimeter on the ceiling but at least at a distance of 1m from the nearest wall and return on any of the walls. 10 linear/slot diffusers were selected to be place at the perimeter of the ceiling on three sides and return was provided on the wall of the fourth side. The return diffuser in this case is located on the east wall, above the door at a height of 3 m with area equal to that in previous models. This specific location was chosen because it will direct the infiltration coming from door towards the return because of the negative pressure, not allowing it to mix in the space. Volume flow rate in each diffuser was 539.8 l/s. Figure 4.6 shows the supply diffusers layout for this configuration and describes the location of return diffuser on the wall in axonometric view. The discharge velocities that were used are 1.5m/s, 2m/s, 3m/s and 3.5m/s similar to previous case.

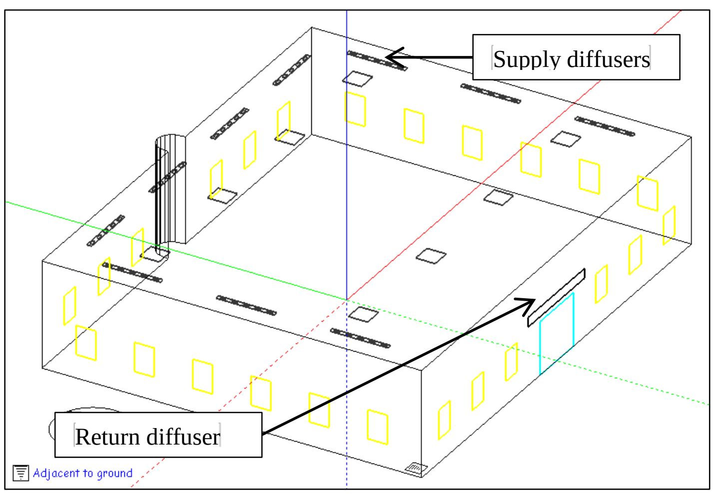


Figure 4.6: M3 Diffuser and return layout

4.2.1.4 CBAD with linear/slot supply diffusers and Wall Return (M3-1)

As a sensitivity analysis of M3 this model was created by changing the location of return diffuser which was brought down to a height of 0.5 m from ground and two return diffusers were used instead of one. The supply diffuser layout was kept same as in M3 and only one diffuser discharge velocity which is 3.5 m/s (most effective) was used. Figure 4.7 axonometric view of the M3-1 model layout.

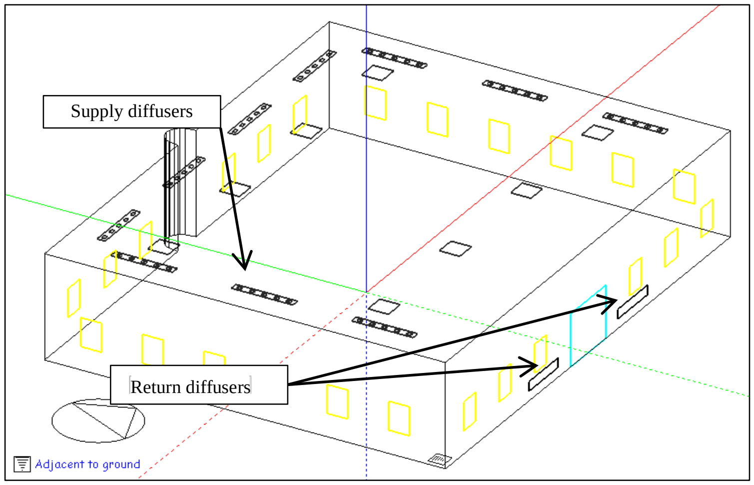


Figure 4.7: M3-1- axonometric view of the layout

4.2.2 Through-Wall Air Distribution (TWAD) (M4 and M5)

In previous air distribution strategies there was a balance between buoyancy and convective effect but if a system predominant with convective effect is used then how the thermal comfort would vary was the question. So Through-Wall Air Distribution was used. This type of air distribution strategy is commonly used when a space has a high ceiling height, such as a mosque building. This strategy is achieved simply by providing supply diffusers at a height of mostly 3 m from ground on the perimeter walls and diffuser type used are liner/slot diffusers. There are two types of diffuser layout configurations used in this system to determine thermal comfort performance of base case. Both will be discussed individually.

4.2.2.1 TWAD with linear/slot supply diffusers and return on Ceiling (M4)

In this configuration, rectangular space of the mosque building was divided into number of rectangles and a diffuser is placed at the center on the shorter side wall in each of these areas. Thus the same concept was used in this case and 8 diffusers were selected with volume flow rate in each diffuser at 674.75 l/s. The return diffusers were provided at the center of the ceiling. Figure 4.8 shows the diffusers layout for this configuration and describes the location of return diffuser at the ceiling in an axonometric view. The discharge velocities that were used are 1.5 m/s, 2 m/s, 3 m/s and 3.5 m/s similar to previous case.

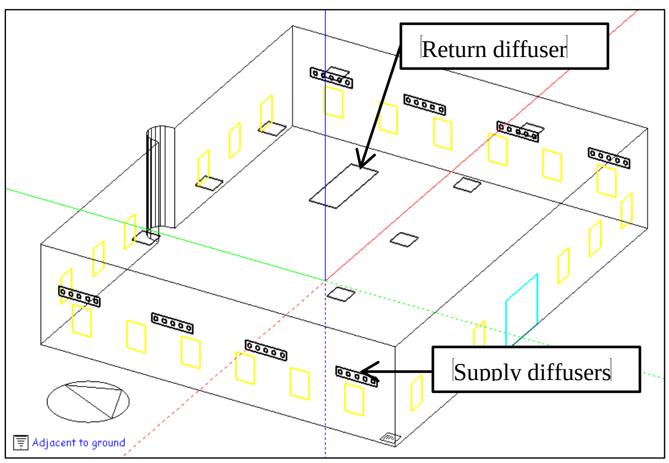


Figure 4.8: M4- axonometric view

4.2.2.2 TWAD with linear/slot supply diffusers and return on wall (M5)

In this configuration, rectangular space of the mosque building was divided into number of rectangles and a diffuser is placed at the center of the perimeter areas. Thus 10 diffusers were selected with volume flow rate in each diffuser at 539.8 l/s. The return diffusers were provided at the center of long wall just above the door at a height of 3m. Figure 4.9 shows the diffusers layout for this configuration and describes the location of return diffuser at the wall in an axonometric view. The discharge velocities that were tested are 1.5m/s, 2m/s, 3m/s and 3.5m/s.

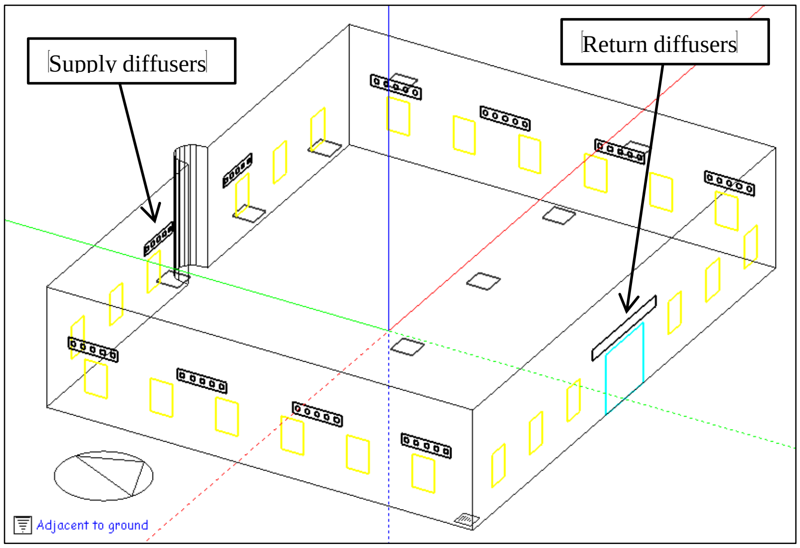


Figure 4.9: M5- axonometric view

4.2.2.3 TWAD with Wall Supply and Ceiling Return (M5-1)

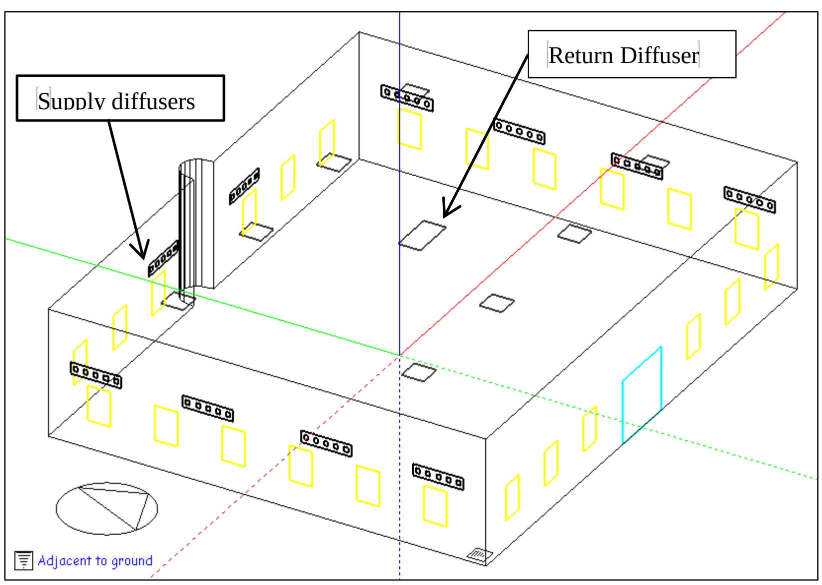


Figure 4.10: M5-1 Layout in axonometric view

M5 required a sensitivity analysis based on the location of return diffuser for intermittent operation. Thus following return sensitivity models were created. The supply locations were kept same as in M5 and the return diffuser was located at the middle of the ceiling. Figure 4.10 shows the diffusers layout for this configuration and describes the location of return diffuser at the ceiling in axonometric view.

4.2.2.4 TWAD with Wall Supply and Ceiling Return (M5-2)

Again the supply locations were kept same and the return diffuser was located at the ceiling almost above the entrance door of the mosque building. Figure 4.11 shows the diffusers layout for this configuration and describes the location of return diffuser at the ceiling in axonometric view.

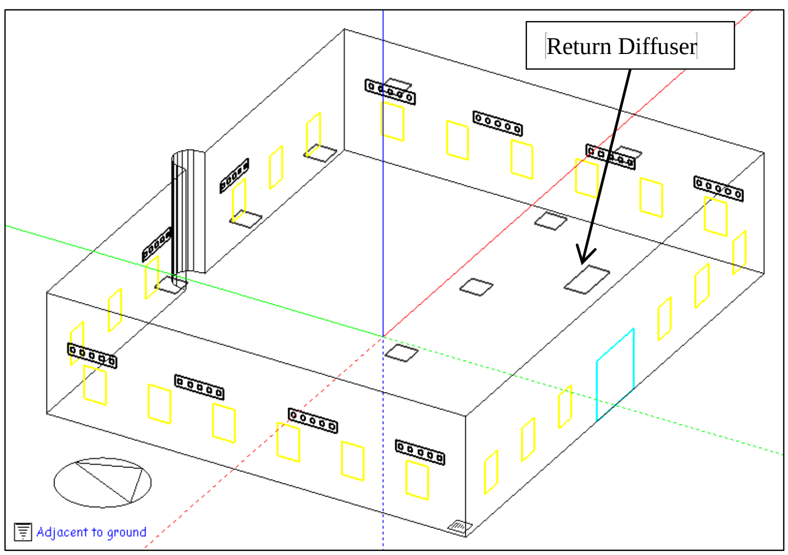


Figure 4.11: M5-2 Layout in axonometric view

4.2.2.5 TWAD with Wall Supply and Wall Return (M5-3)

Again the supply locations were kept same and tow the return diffusers were provided at a height of 0.5 from the ground at both sides of the entrance door of the mosque building.

Figure 4.12 shows the diffusers layout for this configuration and describes the location of return diffuser at the wall in axonometric view.

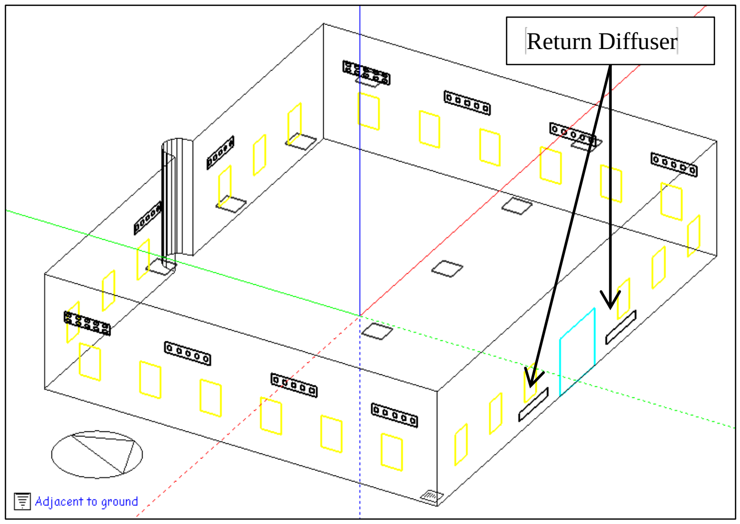


Figure 4.12: M5-3 Layout in axonometric view

4.2.3: Under-Floor Air Distribution (UFAD) (M6 and M7)

This type of air distribution strategy is relatively new and has seen its acceptance in commercial buildings where ceiling height is high. UFAD is known for its temperature stratification purpose which in a large volume space is an effective way for energy conservation but in the occupied zone stratification should not exceed a value of 3^{o}C. UFAD predominantly uses buoyancy effect for its function thus requiring very low velocities compared to other systems. This strategy is achieved simply by providing supply diffusers on ducts raised from floor. Usually the ducts are raised to a height of 1m and scattered uniformly, but for a mosque building it is not possible to provide ducts in the middle of the space. Thus a perimeter layout was opted.

4.2.3.1 UFAD with linear/slot supply diffusers and return on Ceiling (M6)

In this configuration the raised ducts were opted at the perimeter and 16 supply diffuser locations were considered with each supplying a volume flow rate of 337.375 l/s. The return diffuser was provided at the center of the ceiling. Figure 4.13 shows diffuser specification. Figure 4.14 shows the complete configuration in an axonometric view and describes the location of return diffuser at the ceiling. The discharge velocities that were used are 0.8m/s, 1 m/s, 1.25m/s and 1.5m/s because these supply diffusers were very near to occupants and any higher velocities would lead to local velocity drafts.

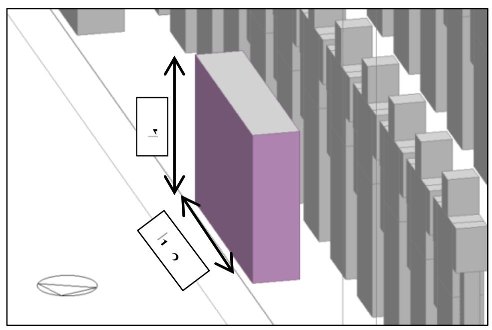


Figure 4.13: M6- Diffuser specification

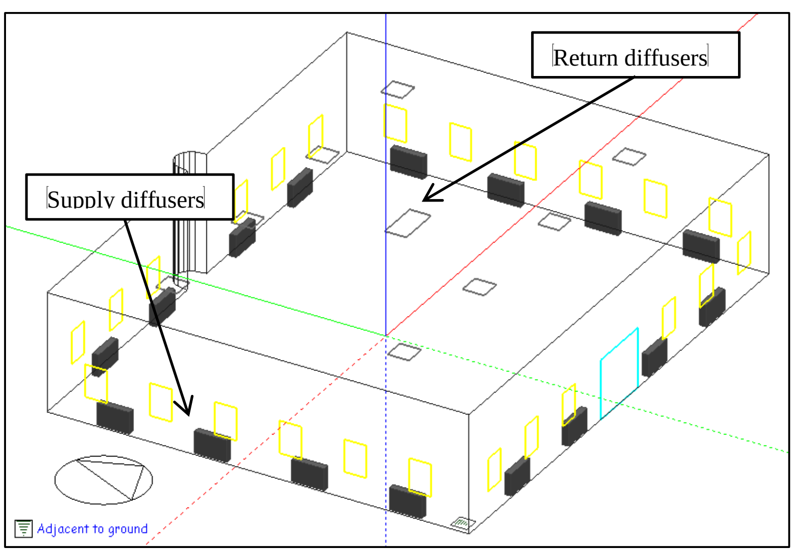


Figure 4.14:M6- configuration axonometric view

4.2.3.2 UFAD with linear/slot supply diffusers and return on Wall (M6)

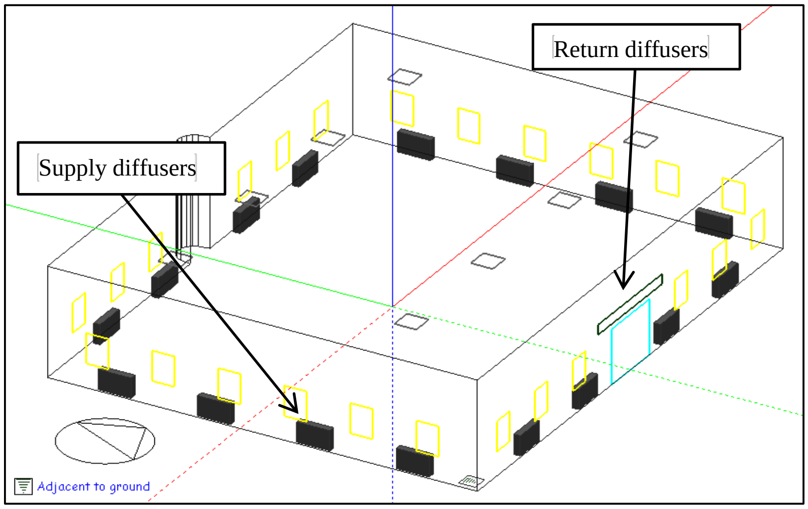


Figure 4.15:M7- configuration axonometric view

In this configuration the raised ducts were located same as previous case with each supplying a volume flow rate of 337.375 l/s. The return diffuser was provided on wall above door at a height of 3m. Figure 4.15 shows the diffusers layout for this configuration in an axonometric view and describes the location of return diffuser at the ceiling. Again the same discharge velocities were used (0.8m/s, 1m/s, 1.25m/s and 1.5m/s).

4.2.3.3 UFAD with linear/slot supply diffusers and return on Wall (M7-1)

As a part of sensitivity analysis of M7, a new model was constructed with same diffuser layout of both supply and return as in M7 but the height of the raised supply duct was increased to 2 m instead of 1 m and simulated for only 1.5 m/s diffuser discharge velocity.

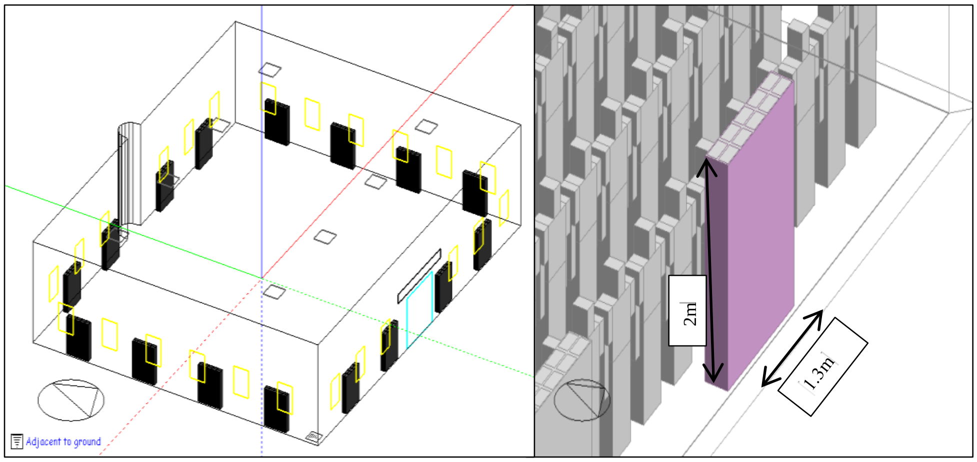
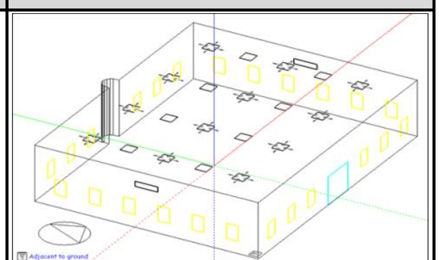
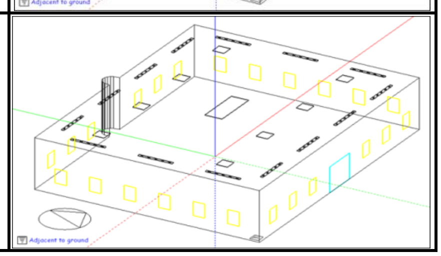


Figure 4.16: M9- Diffuser layout and Supply Diffuser Specification

4.2.4 Summary of air distribution schemes created

Table 4.1: Summary of ceiling based air distribution models

Type	Model No.	Model Description	Velocities Range	Image
CBAD	M1	Four-way ceiling supply and Wall return	1.5 – 3.5 m/s	
	M2	Slot/Linear ceiling supply and ceiling return	1.5 – 3.5 m/s	

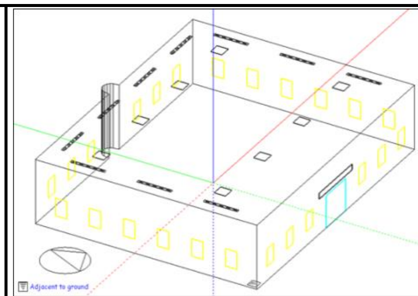
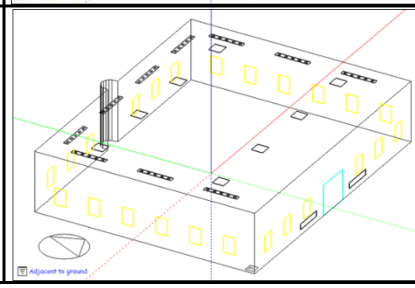
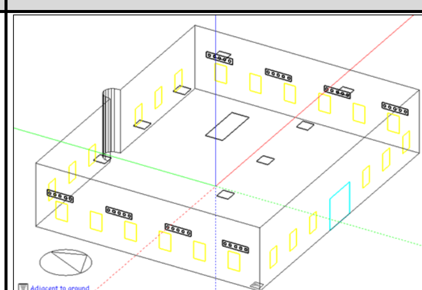
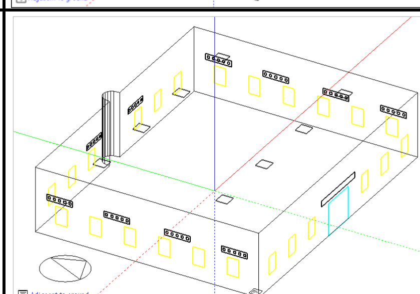
	M3	Slot/Linear ceiling supply and wall return at 3m from ground	1.5 – 3.5 m/s	
	M3-1	Slot/Linear ceiling supply and wall return at 0.5m from ground	3.5 m/s	

Table 4.2: Summary of through wall air distribution models

Type	Model No.	Model Description	Velocities Range	Image
TWAD	M4	Through-Wall supply with 8 diffusers and ceiling return	1.5 – 3.5 m/s	
	M5	Through-Wall supply with 10 diffusers and wall return at 3m from ground	1.5 – 3.5 m/s	

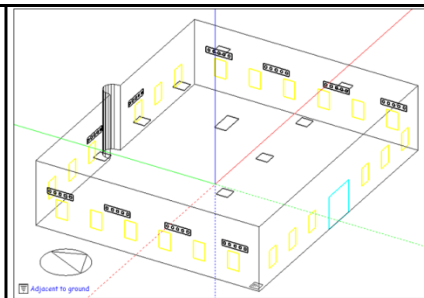
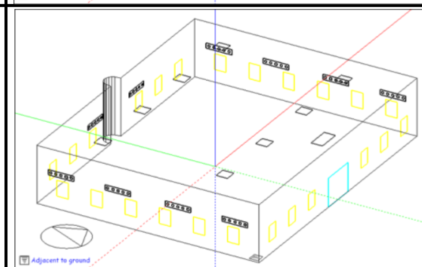
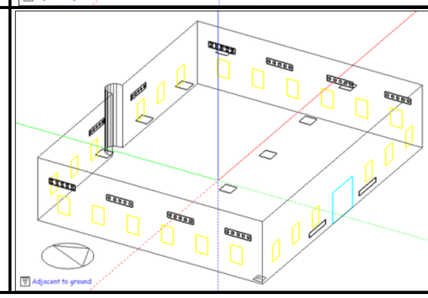
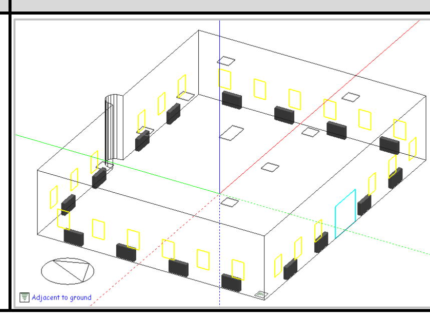
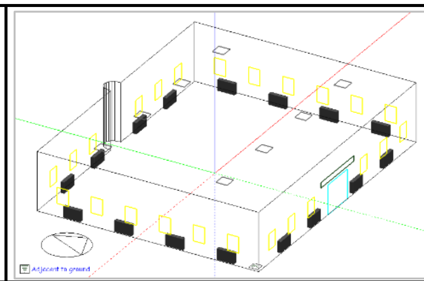
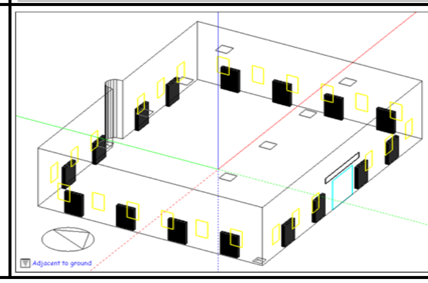
	M5-1	Through-Wall supply with 10 diffusers and ceiling return at center	3.5 m/s	
	M5-2	Through-Wall supply with 10 diffusers and ceiling return near door	3.5 m/s	
	M5-3	Through-Wall supply with 10 diffusers and wall return at 3m from ground	3.5 m/s	

Table 4.3: Summary of under-floor air distribution models

Type	Model No.	Model Description	Velocities Range	Image
UFAD	M6	Under-floor supply and ceiling return	0.8 – 1.5 m/s	

	M7	Under-floor supply and wall return at 3m from ground	0.8 – 1.5 m/s	
	M7-1	Under-floor supply with duct size 2 m and wall return at 3m from ground	1.5 m/s	

To summarize, there were 12 models created for CFD analysis and Table 10 shows the model number, description and velocities range for each created model. In the next chapter only model numbers will be used when displaying and analyzing the results of each of the models.

4.3 CFD Meshing

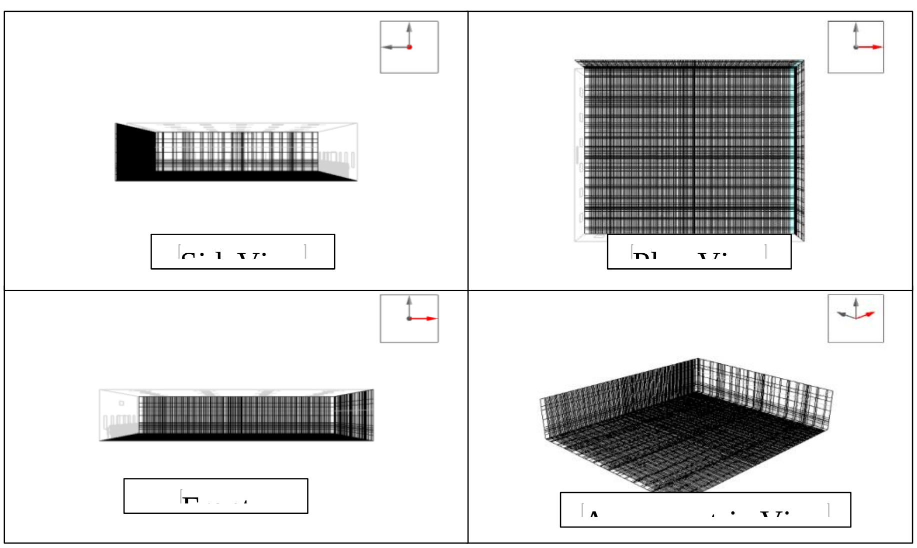


Figure 4.17: Created Mesh in different view angles

The next in CFD calculation method is the division of the geometric space across which the calculations are to be conducted into a number of non-overlapping adjoining finite volume grid or cells. In DesignBuilder when a CFD project is created, a grid is automatically generated for the required model domain by identifying all contained model object vertices and then generating key coordinates from these vertices along the major grid axes. These key coordinates which are known as ‘grid lines’, extended from the X, Y and Z-axes across the width, depth and height of the domain respectively. In this study the default grid spacing which is the region between grid lines along each axis, was considered to be 0.6 m which is equivalent to shoulder broadness of an average human. In DesignBuilder very narrow grid regions are avoidable by merging very close adjacent grid lines formed from key coordinates together using the merge tolerance setting in order to reduce the aspect ratio. A value of 0.06 m was used as merge tolerance setting in most simulations that gave an aspect ratio of 17, where the allowable value is 26. And in some cases a value of 0.1 m was used when aspect ratio was higher than 17 to reduce it to 17 or below. Figure 4.7 displays different views of the mesh that was created for CBAD that used four way diffusers (M1). On average there were 300,000 cells created.

CHAPTER 5

ANALYSIS, DISCUSSION AND RESULTS

The initial phase of the current chapter focuses on the base case model, which has been developed earlier and investigates its performance for thermal comfort and energy end-use. The EnergyPlus engine is used to assess energy performance of different HVAC operation strategies. This is followed by the analysis of thermal comfort status at the occupant level for different air distribution schemes using CFD and Fanger thermal comfort module of DesignBuilder. Finally, the chapter concludes by discussing and evaluating the resulting combined energy performance and thermal comfort status for the different operation strategies and air distribution schemes.

5.1 Energy and Thermal comfort Analysis of Base Case

5.1.1 Energy Performance

The verified base case model of mosque building was simulated using state-of-the-art DesignBuilder simulation program that uses EnergyPlus engine for annual simulation using the weather data file of Dhahran for the year 2012 for continuous operation of HVAC system. Results of total building energy consumption for each month are shown in Figure 5.1 and monthly cooling energy consumption is shown in **Error! Reference source not found.**. The consumption for the base model was 85218.5 kWh or 181.62 kWh/m^2 of which 157.80 kWh/m^2 going for cooling alone representing about 87% of the total energy consumption. The month of August recorded the highest monthly energy consumption with 15142.1 kWh or 32.27 kWh/m^2 and February Recorded the lowest at

1164.50 kWh or 2.48 kWh/m^2 which is reasonable based on the weather conditions of Dhahran in Kingdom of Saudi Arabia. The overall energy consumption can be considered to be the high side because of the HVAC System which in this case being operated continuously.

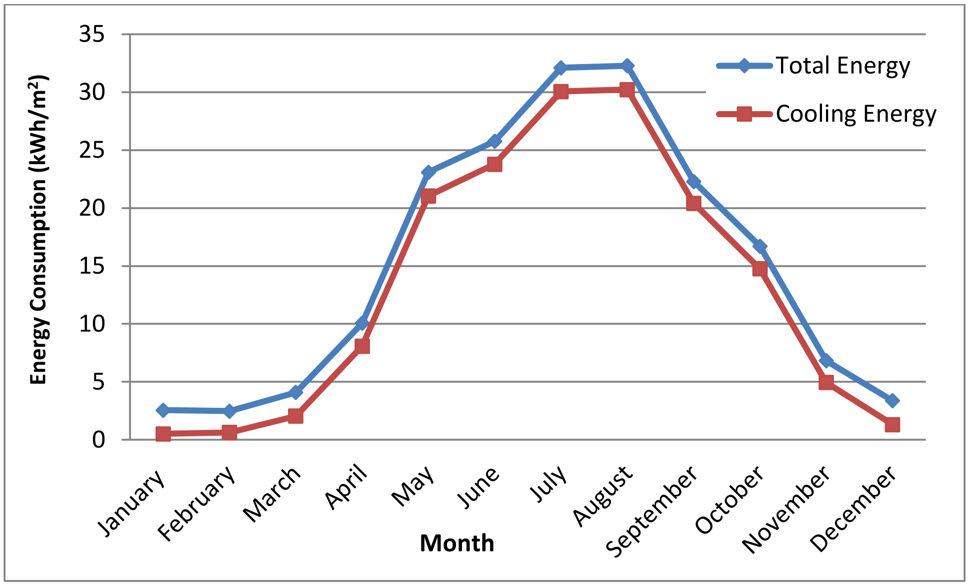


Figure 5.1: Monthly Total and Cooling Energy consumption

5.1.2 Thermal Comfort Performance

A space is said to be thermally comfortable for its occupants when the occupied zone which extends from ground level to a height of 2m across the space, is within thermal comfort requirements. For a space to be comfortable the PMV should be between -0.5 to 0.5 which would be an ideal situation but it is very difficult to achieve the whole space within these values. ASHRAE standard 55 requires that 80% of the occupants be satisfied with thermal environment of the space, thus a variation between -1 and 1 value of PMV throughout a space would be acceptable. The corresponding air temperature values are 21 ^{o}C to 26 ^{o}C with acceptable velocity of 0.10 to 0.8m/s in the occupied zone for operative temperature values of 22 ^{o}C to 29 ^{o}C would fall under the acceptable PMV values. Temperature offset will occur in locations where air velocity is above 0.2 m/s and maximum allowed temperature offset value is 3^{o}C. The temperature offset will be discussed based on Figure 2.1. Predicted Percentage Dissatisfied (PPD) values corresponding to different PMV values are again presented here to show how the required value of 80% is achieved:

I. $-0.2 < PMV < +0.2$ ($PPD \leq 6\%$)
II. $-0.5 < PMV < +0.5$ ($PPD \leq 10\%$)
III. $-0.7 < PMV < +0.7$ ($PPD \leq 15\%$)
IV. $-1.0 < PMV < +1.0$ ($PPD < 30\%$)

Results of thermal comfort status on 21st July which present a typical summer design day are presented in Figure 5.2. It can be seen that the average PMV over the space varies between -0.1 and 0.44 throughout the day which corresponds to PPD of 10% that is well within ASHRAE standard 55 limits. The MRT value calculated by the software is considered acceptable when temperatures of different surfaces within the space don't vary greatly as the MTR is angle dependent factor. The corresponding average MRT that the software calculates is considered reliable since the EnergyPlus uses the concept of heat balance. Average MRT for this simulation varied between 27^{o}C to 28^{o}C throughout the day. It should be noted however, that the software assumes a uniform velocity of 0.13 m/s and air temperature, which is the set point temperature for the simulation that is 24^{o}C in this case, to be uniform throughout the space. In reality, this uniformity of values is very difficult or impossible to achieve in large spaces as the velocity and temperature vary in the space depending upon diffuser location, discharge velocity of diffuser, load distribution, location of the return diffuser etc. So the PMV values that the software predicted need to be verified by simulating an environment close to reality using CFD technique. For the impact of different air distribution schemes and different values of diffuser discharge velocity on status of thermal comfort need to be investigated.

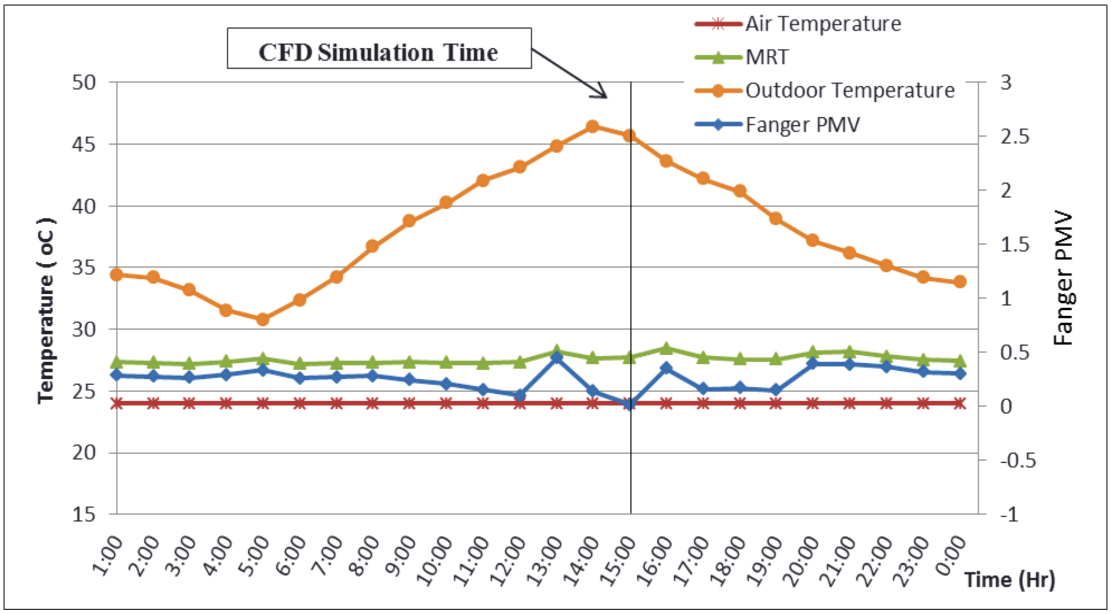


Figure 5.2: Thermal Comfort status on 21st July for the modelled mosque

5.2 Assessment of Thermal Comfort with continuous HVAC operation

This part analyses and discusses the impact of different air distribution schemes on thermal comfort status when the HVAC system is working continuously and presents the results obtained. All the models created for CFD analysis were simulated and then analyzed for results. The results are presented and discussed in two contour sections that were taken at height of 0.3 m and 1.5 m from the ground as the occupants in mosques are not only standing but for a good portion of the time are seated on ground as well. So these two sections would provide a good picture of the thermal comfort status of the occupied zone. Contour presents the variation of different parameters in the space around the occupant assemblies that are shown in rows as the contours are presented in plan/top view. Figure 5.3 shows the axonometric view of the two sections. The results are discussed based on four parameters of thermal comfort namely Air Velocity, Operative Temperature, PMV and PPD.

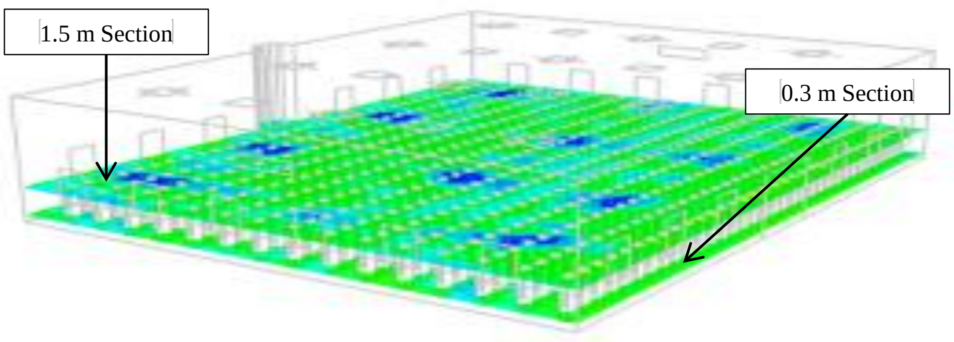


Figure 5.3: 0.3 m and 1.5 m contours section from close side view

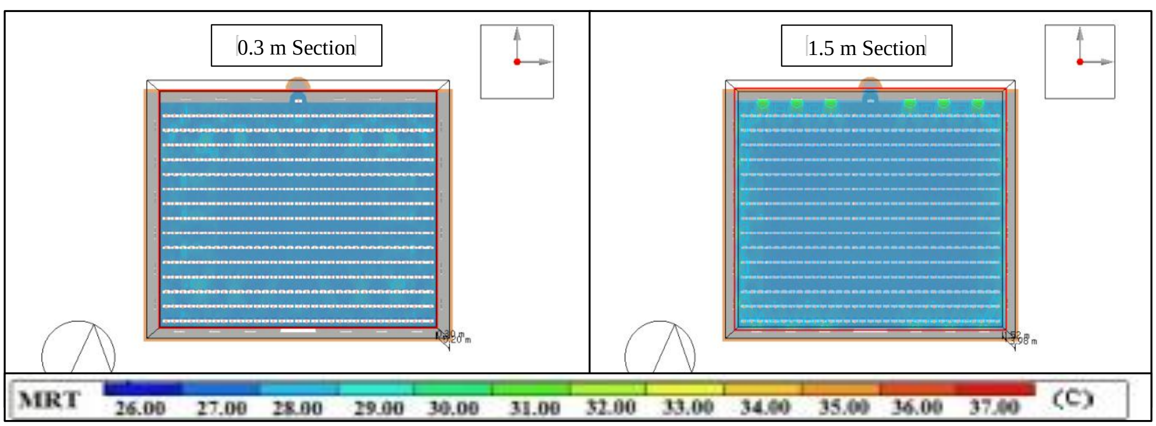


Figure 5.4: MRT contours for section 0.3 m and 1.5 m

As the surface temperature boundary conditions and the setup remain same for all the CFD simulations the resultant MRT would also be same for all simulations. The MRT contours for 0.3 m and 1.5 m sections are presented in Figure 5.4. Mean Radiant Temperature (MRT) values were observed to be around 27^{o}C to 28^{o}C for both sections, except near west facing windows (1.5m section) where MRT was around 31^{o}C which is reasonable, as this side is Sun-facing at simulation time making window temperatures to be higher. The average value was almost similar to the EnergyPlus prediction.

5.2.1 Ceiling-Based Air Distribution (CBAD) (M1, M2 and M3)

5.2.1.1 CBAD with four way supply diffusers (M1)

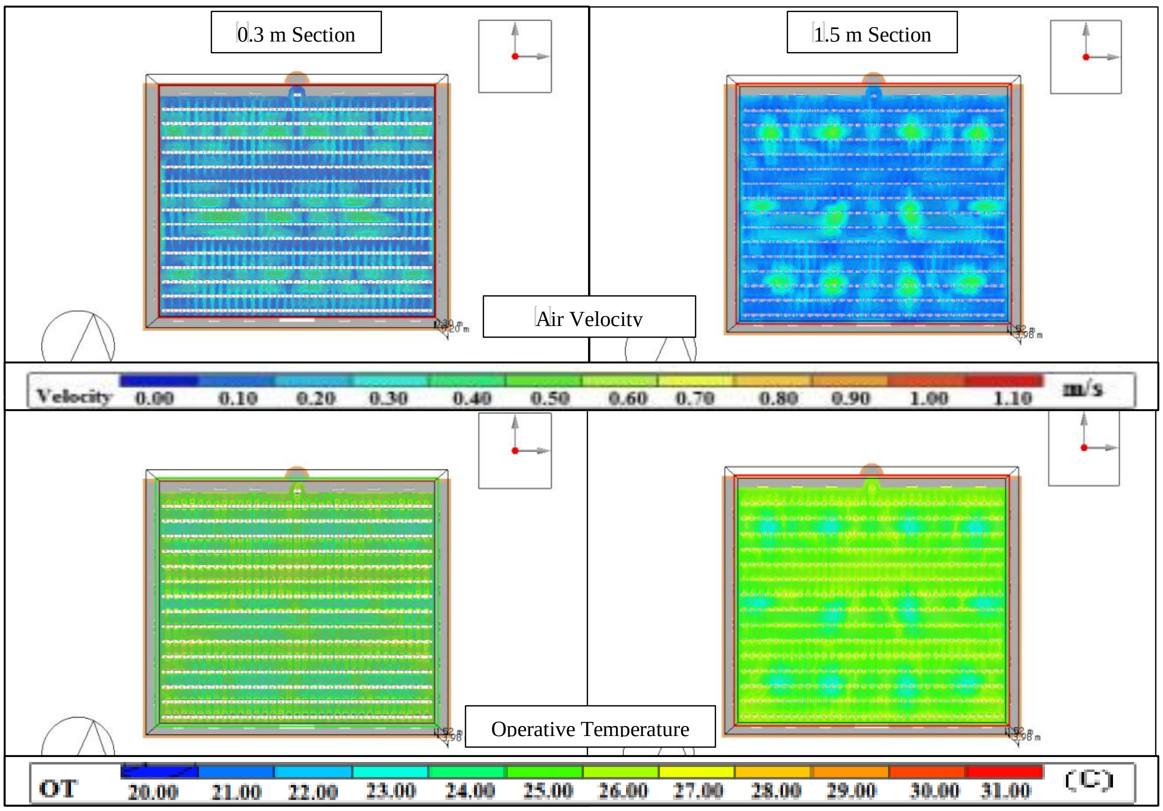


Figure 5.5: M1- Air velocity and Operative Temperature contours at sections 0.3m and 1.5m for 1.5m/s velocity

M1 air distribution scheme was simulated for thermal comfort performance. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.5 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.6 for a diffuser discharge velocity of 1.5m/s. Air velocity was found to be varying between 0.1m/s to 0.5m/s in both sections with velocity above 0.2m/s usually occurring exactly below the diffuser location. This means that there will be temperature offset occurring in locations where velocity is above 0.2 m/s up to a value of 2°C. Operative Temperature which is essentially an average of air temperature and MRT varied between 22°C to 26°C mainly influenced by air temperature and is within specified limits in both sections with 25°C being predominant. Operative Temperature variation of 22- 24°C was mostly observed to occur exactly below the diffuser locations although these values of operative temperature are within allowable limits.

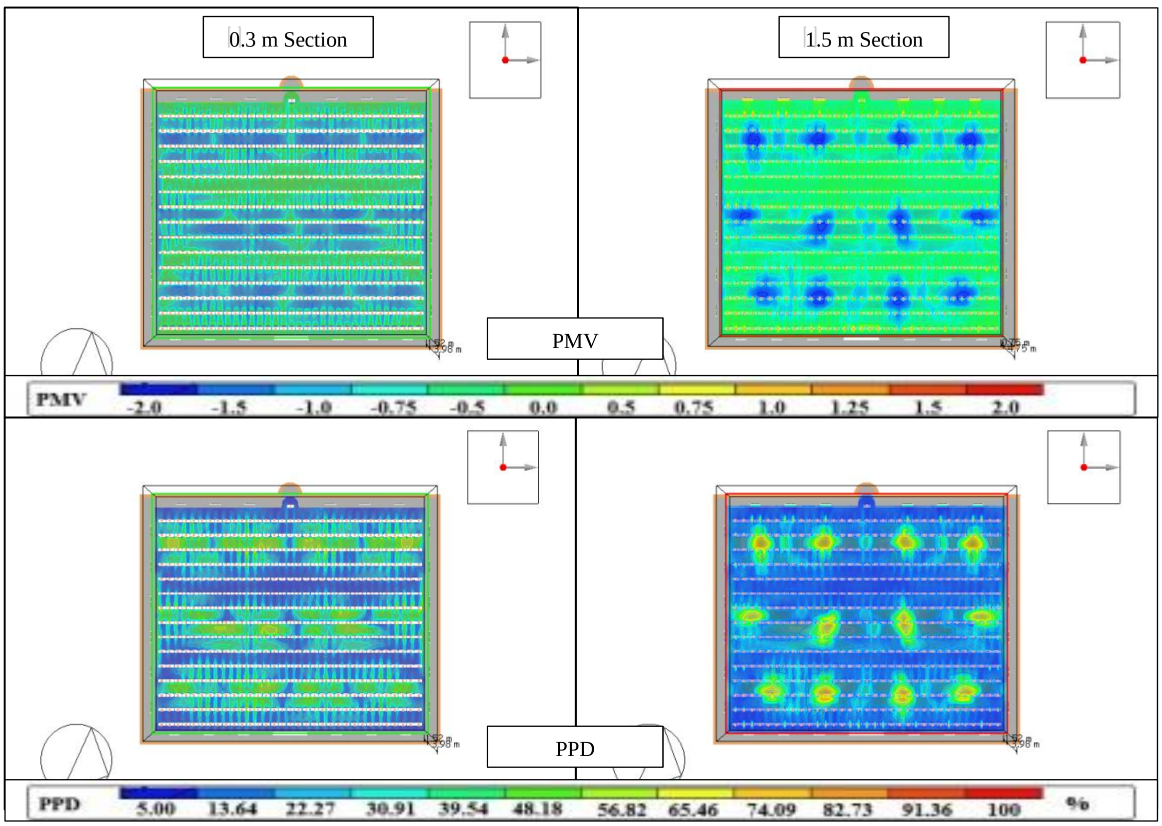


Figure 5.6: M1- PMV and PPD contours at sections 0.3m and 1.5m for 1.5m/s velocity

PMV was found to vary from -1.5 to 0.0 and -1.5 to -0.5 PMV in sections 0.3m and 1.5m respectively. Lower values were found to occur at places where velocity was above

0.2m/s and operative temperatures below 24°C. This lower PMV values are not desirable as most people will feel uncomfortable in these regions.

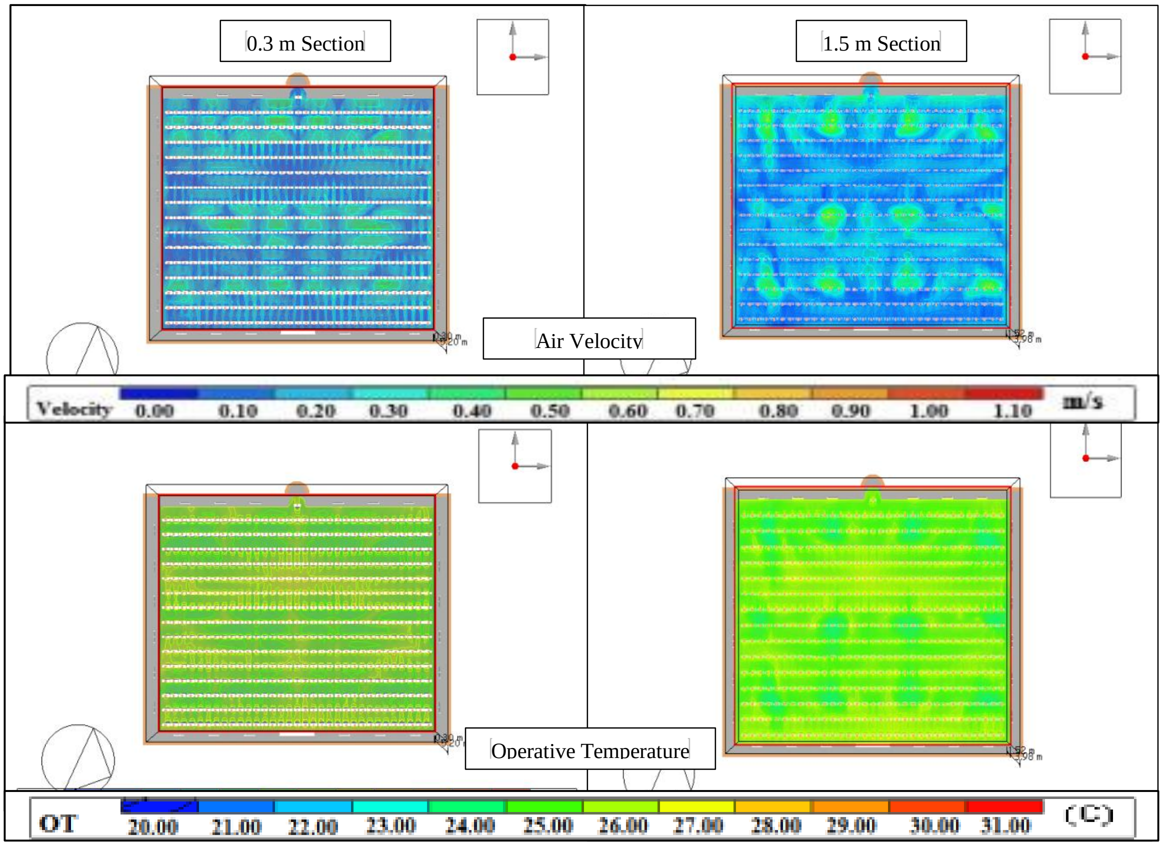


Figure 5.7: M1- Air velocity and Operative Temperature contours at sections 0.3m and 1.5m for 3.5m/s velocity

The resultant PPD for both sections was found to vary from 5% to 82% with higher value was observed in the region of drafts. For this diffuser discharge velocity the space can be termed as thermally not comfortable as there are cold spots which are uncomfortable areas for most occupants. Even though the diffusers are discharging air in four different directions in order to mix it uniformly, the buoyancy force dominated the flow and forced the air to not mix properly resulting in cold spots. In order to improve the thermal comfort situation, either the number of diffusers needs to be increased which is not desirable economically and aesthetically or diffusers with higher discharge velocity need to be used in order to arrive at a better situation. Thus a diffuser discharge velocity of 2 m/s was used and results were analyzed. However, results of 2 m/s discharge velocity showed not much difference compared to previous velocity case. The cold spots still existed with almost the same values of all parameters which prompted the use of 3 m/s as diffuser discharge velocity.

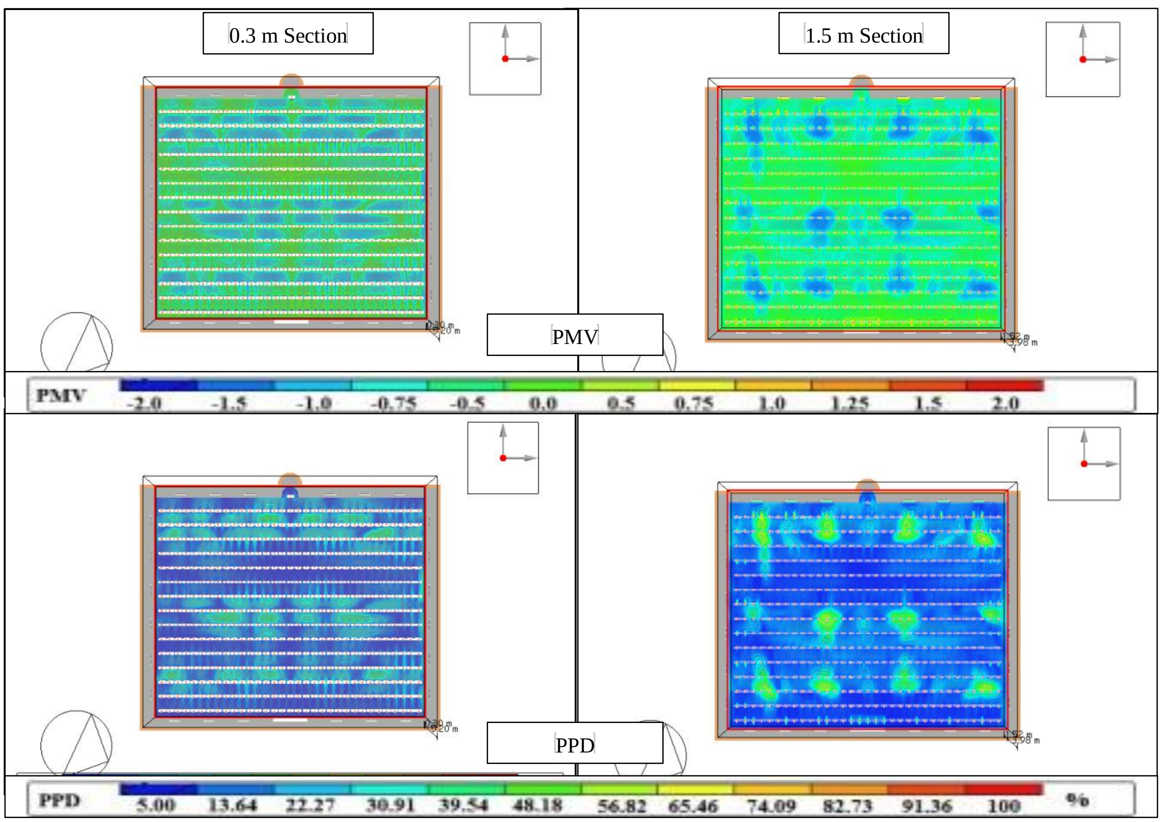


Figure 5.8: M1- PMV and PPD contours at sections 0.3m and 1.5m for 3.5m/s velocity

This increase in velocity showed much better results compared to previous cases. It was observed that effectiveness of the diffusers increased with this velocity to mix the air to more uniform temperature but still cold areas existed in the occupied zone due to elevated unwanted air movement which resulted in drafts. The resultant PPD for 0.3m section was found to vary from 5% to around 50% in the regions of draft, and for 1.5m section, was found to vary from 5% to around 74% in the regions of draft which encouraged the using of higher discharge velocity in order to arrive at a better situation. Lastly a diffuser discharge velocity of 3.5 m/s was used. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.7 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.8 for a diffuser discharge velocity of 3.5m/s. Air velocity for 0.3m section was found to be varying between 0.1m/s to 0.4 m/s with velocity above 0.2m/s usually occurring exactly below the diffuser location and air temperature offset caused is less than 2^{o}C as the temperature in these areas is around 23^{o}C. But for 1.5m section it was found to be varying between 0.1m/s to 0.5m/s with velocity above 0.2m/s mostly occurring exactly below the diffuser location and air temperature offset caused is again less than 2^{o}C as air temperature in these areas. Operative Temperature varied between 24^{o}C to 26^{o}C with 25^{o}C being predominant which again is within specified limits. PMV was in the range of -1.0 to 0.0 in 0.3m section, which is an improved situation and for 1.5m section, PMV below diffuser locations was -1.25 and in other region, between -1 and 0.0. It was observed that effectiveness of the diffusers increased further with this velocity to mix the air to more uniform temperature and very little cold spots exist in the occupied zone that is usually caused by elevated unwanted air movement which resulted in drafts. The resultant PPD for 0.3m section was found to vary from 5% to around 40% with 40% occurring in the regions of draft and for 1.5m section, PPD was found to vary from 5% to around 50% with higher value occurring in the regions of draft. For this diffuser discharge velocity the space can be termed as thermally not comfortable but overall situation improved compared to the previous velocity cases. The reason for four-way diffusers which are known for uniform mixing of air, to be ineffective is the ceiling height. For the given base case model height, a further increase in the velocity would have yielded better results but it is not practical to use such high diffuser discharge velocity which can cause unwanted noise. This diffuser configuration works towards achieving comfort by utilizing forced convection method and does not utilize the buoyancy effect and negative pressure created towards return air diffusers. It created an interest to test a system that would utilize these two phenomenons and see how the thermal comfort varies.

5.2.1.2 CBAD with linear/slot supply diffusers and Ceiling Return (M2)

M2 air distribution scheme was simulated for thermal comfort performance. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.9 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.10 for a diffuser discharge velocity of 1.5m/s. Air velocity was found to be varying between 0.1m/s to 0.5m/s in both sections with velocity above 0.2m/s mostly occurring exactly below the diffuser locations and extended towards center due to negative pressure of the return. This means that there will be temperature offset occurring in these locations upto a value of 2^{o}C.

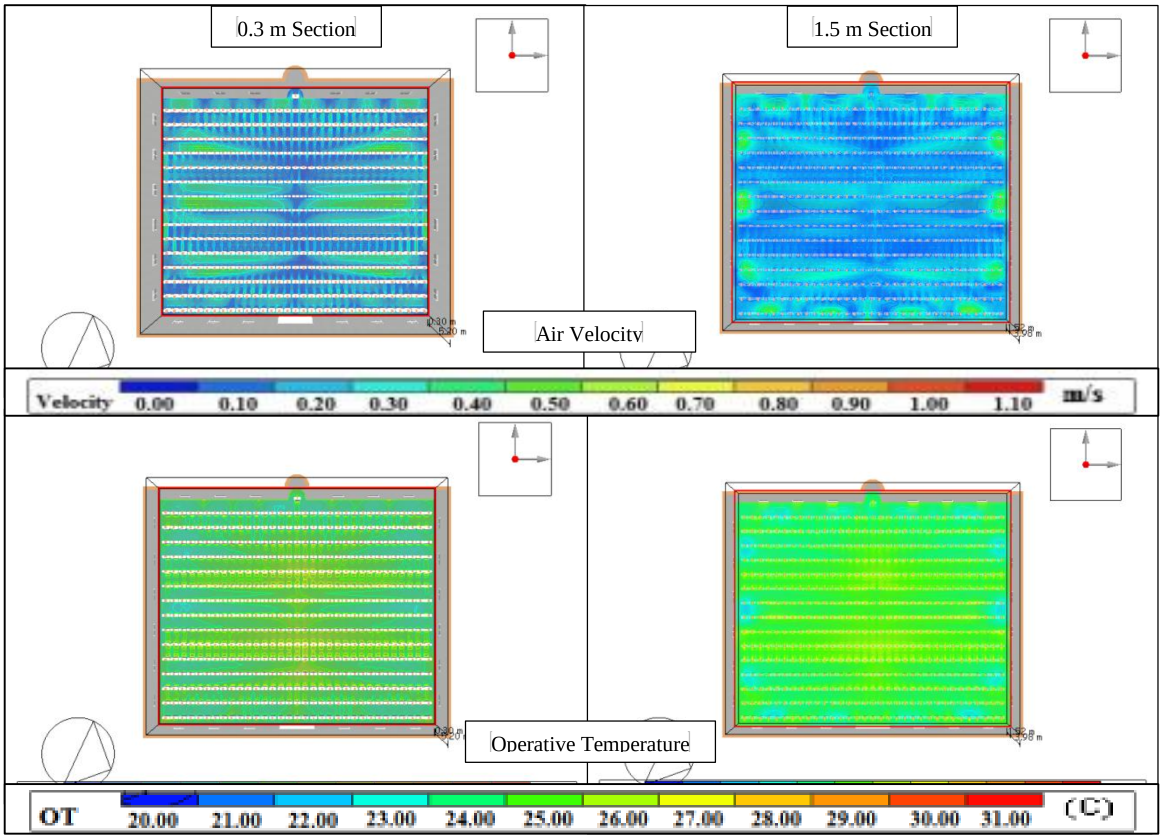


Figure 5.9: M2- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

Operative Temperature was found to vary between 22°C to 26°C mostly tending towards cool side with 23°C being predominant. Operative temperature variation of 22- 23°C was mostly observed to occur exactly below the supply diffuser locations. These values of operative temperature are towards cold side and would form a temperature draft. PMV was found to vary from -1.25 at the perimeter region below the supply diffuser locations to 0.0 in the middle region. PMV values for 0.3 section is towards cool side because of the velocity drafts which extended towards center due to negative drag created towards the return diffuser location. The resultant PPD for both sections was found to vary from 5% to 65% with higher value observed in the region of velocity drafts. For this diffuser discharge velocity, the space can be termed as thermally not comfortable, but the overall situation is improved compared to M1 model with same velocity. These cold spots are mostly due to buoyancy effect and can be reduced by increasing the convection effect which can be done by increasing the supply diffuser discharge velocity.

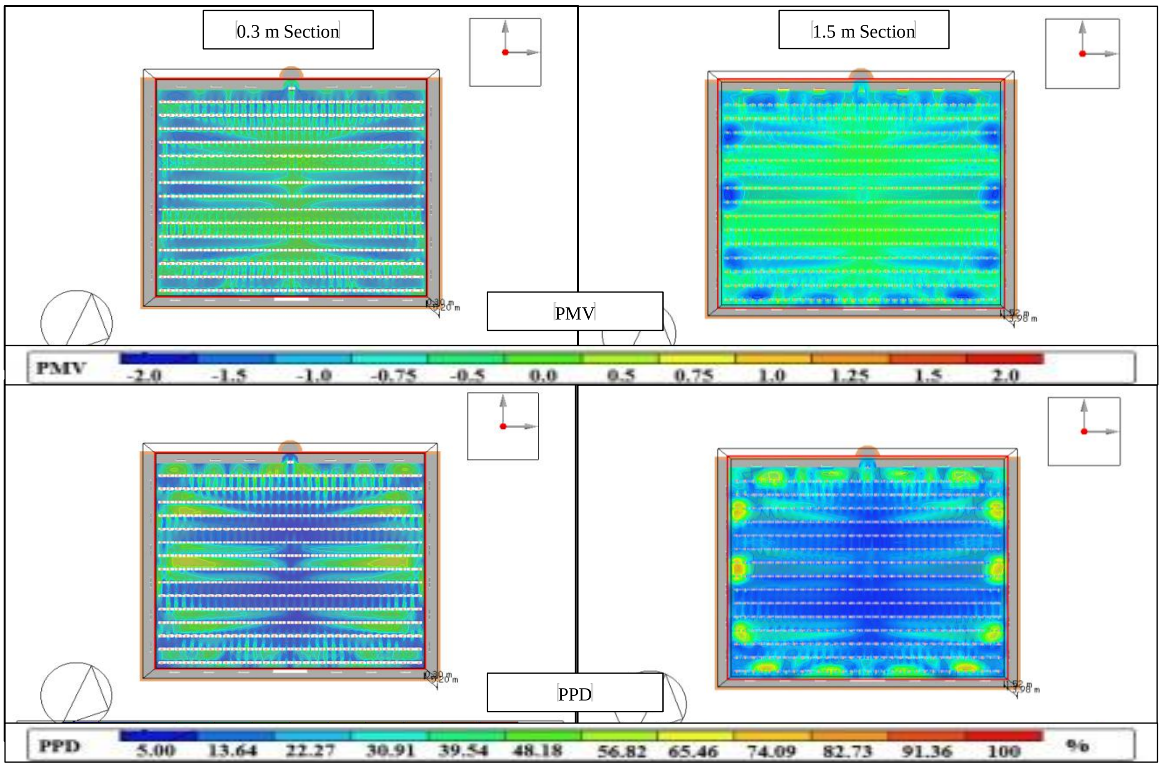


Figure 5.10: M2- PMV and PPD contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

A velocity of 2 m/s was tested in order to assess the improvement in the situation. Similar values for air velocity, and operative temperature were observed with this velocity with PMV varying from -1.25 at the perimeter region to 0.0 in the middle region. The resultant PPD for both sections was found to vary from 5% to 65% in the region of drafts making this setup uncomfortable similar to previous velocity case. The supply diffuser discharge velocity was increased to 3 m/s and results show that air temperature varied between

21^{o}C to 24^{o}C in both sections with 23^{o}C being predominant and again temperature variation of 21- 22^{o}C was mostly observed to occur exactly below the diffuser locations, again resulting in temperature draft. Air velocity contours were found to be varying between 0.1m/s to 0.5m/s with occurrence of velocity drafts. PMV was varying from -1.25 at the perimeter region to 0.0 in the middle region with -0.5 being predominant. The resultant PPD for both sections was found to vary from 5% to 70% which is an improved situation compared to previous case. As the diffuser discharge velocity increased, the space was observed to move towards a comfortable environment but for this velocity case space did not achieve 80% requirement to be termed comfortable. This prompted a further increase in supply diffuser discharge velocity to 3.5m/s. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.11 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.12 for a diffuser discharge velocity of 3.5m/s. Air velocity contours were found to be varying between 0.1 m/s to 0.4 m/s in both sections with 0.25 m/s as average velocity value. In this case temperature offset occurring in the elevated velocity regions was observed to be less than 1.5^{o}C which is the best value so far. Operative Temperature was found varying between 22^{o}C to 26^{o}C in both sections with 24^{o}C being predominant and temperature variation of 22- 24^{o}C was mostly observed to occur exactly below the diffusers. PMV was found to vary from -1.25 at the perimeter region to 0.0 in the middle region with -0.5 being the average value. It was observed that lower PMV values were due the low temperature value but not mainly because of velocity drafts which goes to show the combine effect of buoyancy and convection. The resultant PPD for both sections was found to vary from 5% to 50% with an average of 22%. Although this value is very near to the required 20% or less PPD but for this diffuser discharge velocity the space can be termed as thermally not comfortable. However, M2 performed better than M1 but still fall short of the requirement. This is because the return was on ceiling and again buoyancy faces dominated. So a model with return on wall near to the occupant zone could enhance the situation.

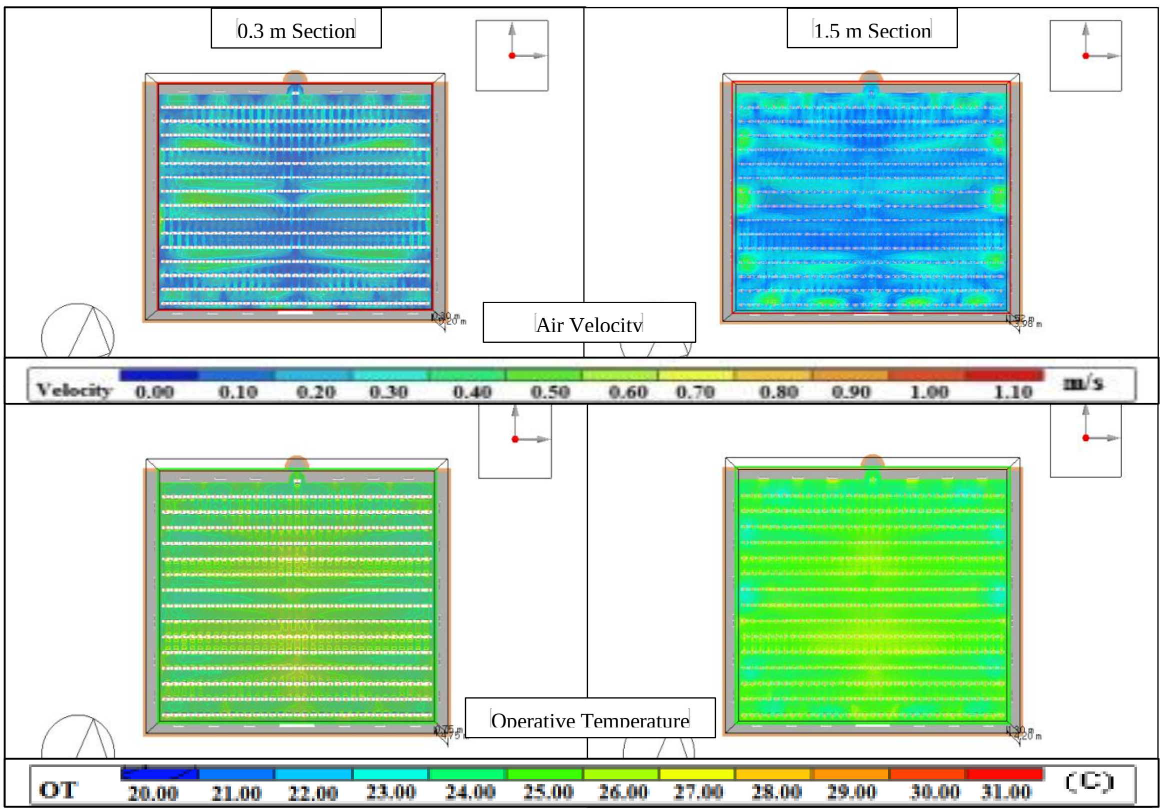


Figure 5.11: M2- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

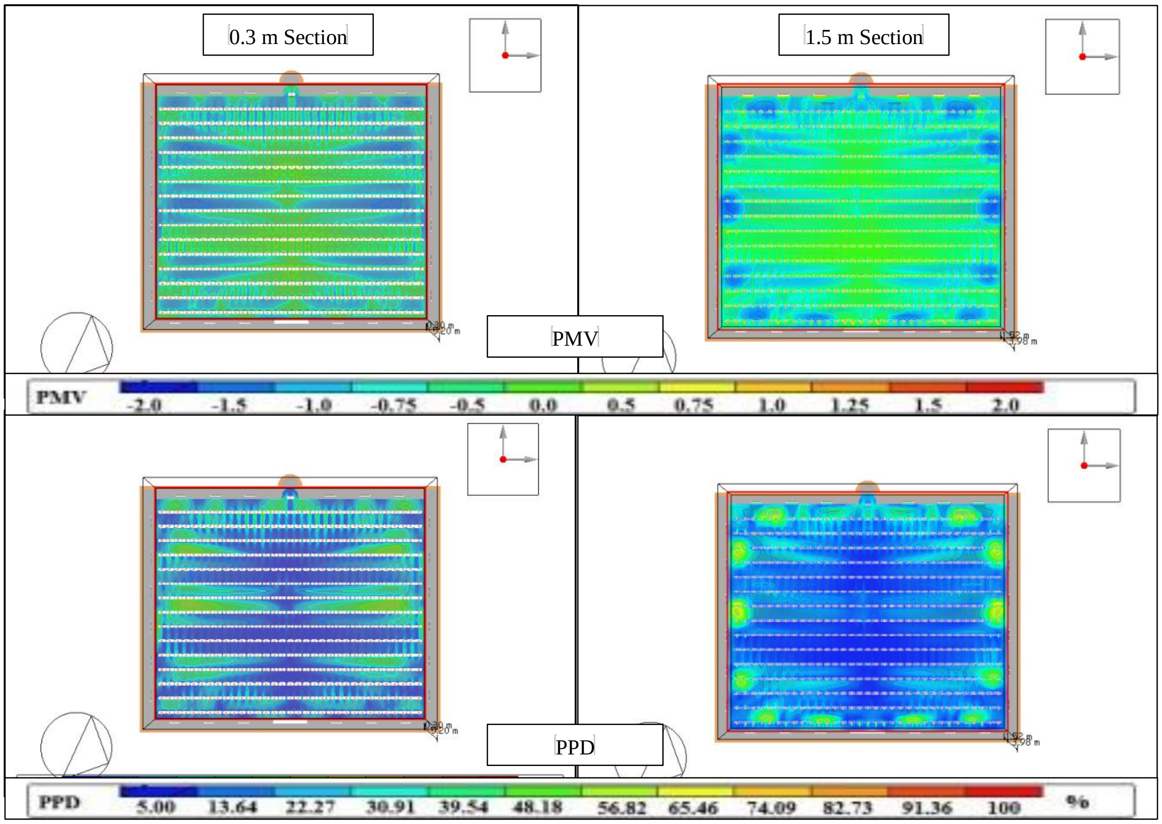


Figure 5.12: M2- PMV and PPD contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

5.2.1.3 CBAD with linear/slot supply diffusers and Wall Return (M3)

M3 air distribution scheme was simulated for thermal comfort performance. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.13 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.14 for a diffuser discharge velocity of 1.5m/s. Air velocity was found to be varying between 0.1m/s to 0.5m/s in both sections with 0.3 m/s velocity the entire occupied zone. Thus the temperature offset occurring in locations with velocity higher than 0.20 m/s was observed to be less than 2^{o}C. Operative Temperature was varying between 22^{o}C to 26^{o}C with very few spots of 22^{o}C below the supply diffuser location which shows the effect of MRT as at these location was MRT higher due to hot surface of the glass windows face sun. This reduced the effect of very low air temperature prevailing in those regions. As a result PMV was found to vary from -1.25 at the perimeter region to -0.5 to 0.0 in the entire occupied zone which is encouraging. The resultant PPD for both sections was found to vary from 5% to 40% in the region below diffusers. It goes on to show that the space is mostly comfortable due to a balance between buoyancy and convection effect, but with temperature and velocity drafts occurring exactly below diffusers locations. The draft regions need to be reduced which was done by increasing the supply diffuser discharge velocity.

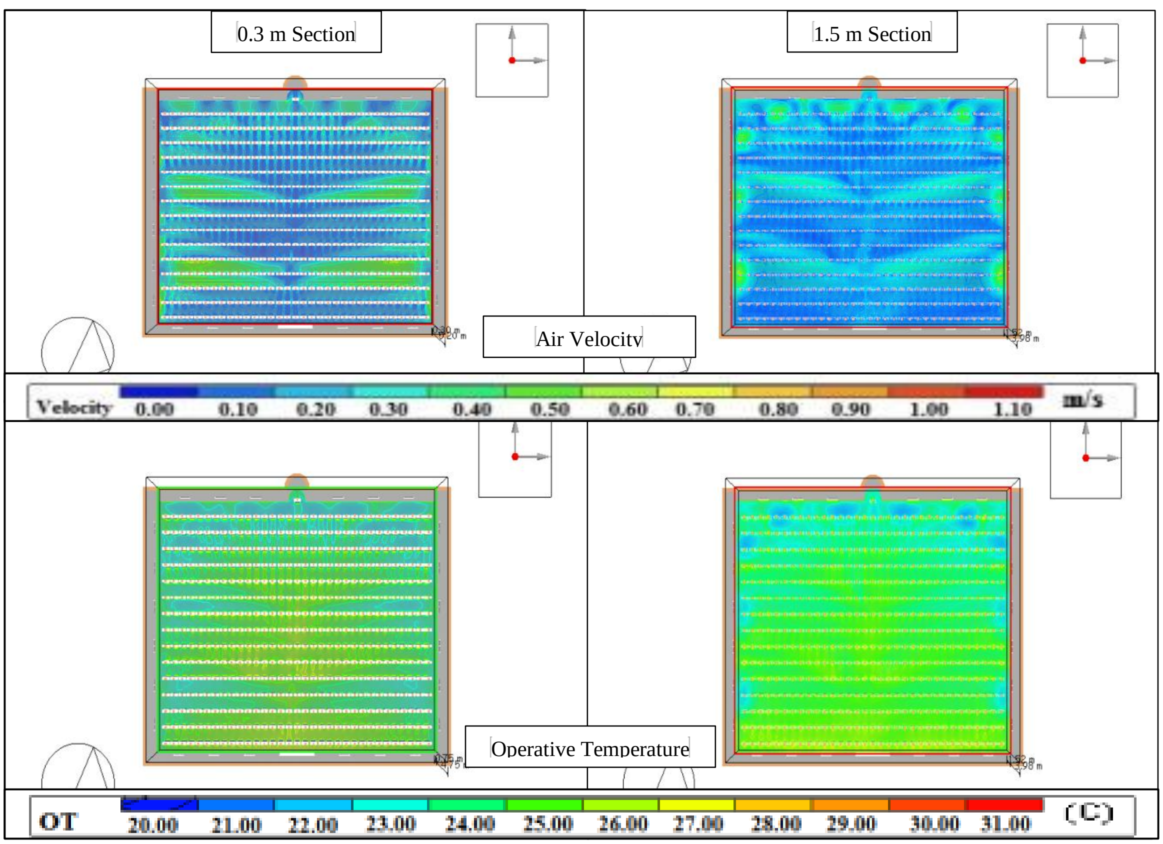


Figure 5.13: M3- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

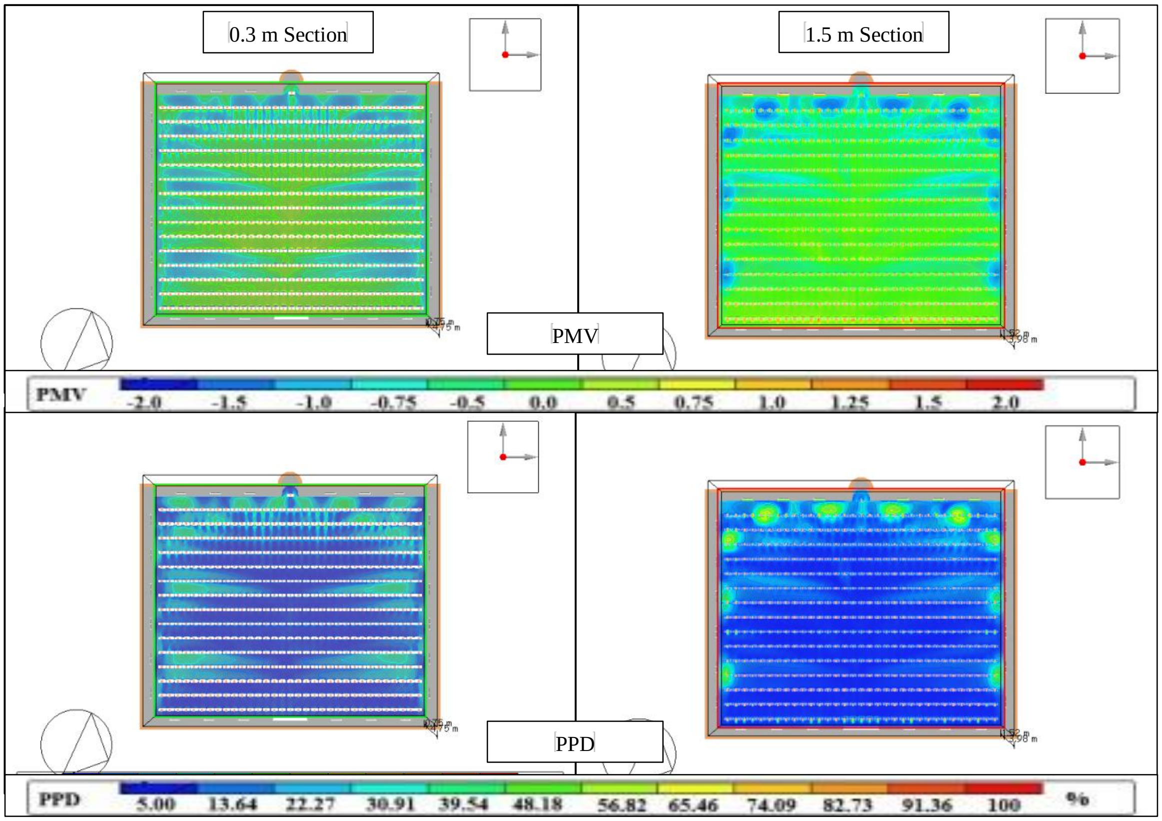


Figure 5.14: M3- PMV and PPD contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

For a diffuser discharge velocity 2 m/s the situation was more or less the same. So supply diffuser discharge velocity was increased to 3 m/s. The comfort regions were more uniform with PMV improving to -1.0 at the perimeter region to -0.5 to 0.0 in the middle region. The resultant PPD for both sections was found to vary from 5% in most regions to 40% in the region below diffusers. This space can be termed thermally comfortable for 80% or higher occupants. It was observed that higher diffuser discharge velocity improved thermal comfort and reduced the draft regions, by increasing the balance between buoyancy and convection effects. Lastly a diffuser discharge velocity of 3.5 m/s was used to ensure more reduction of cold spots.

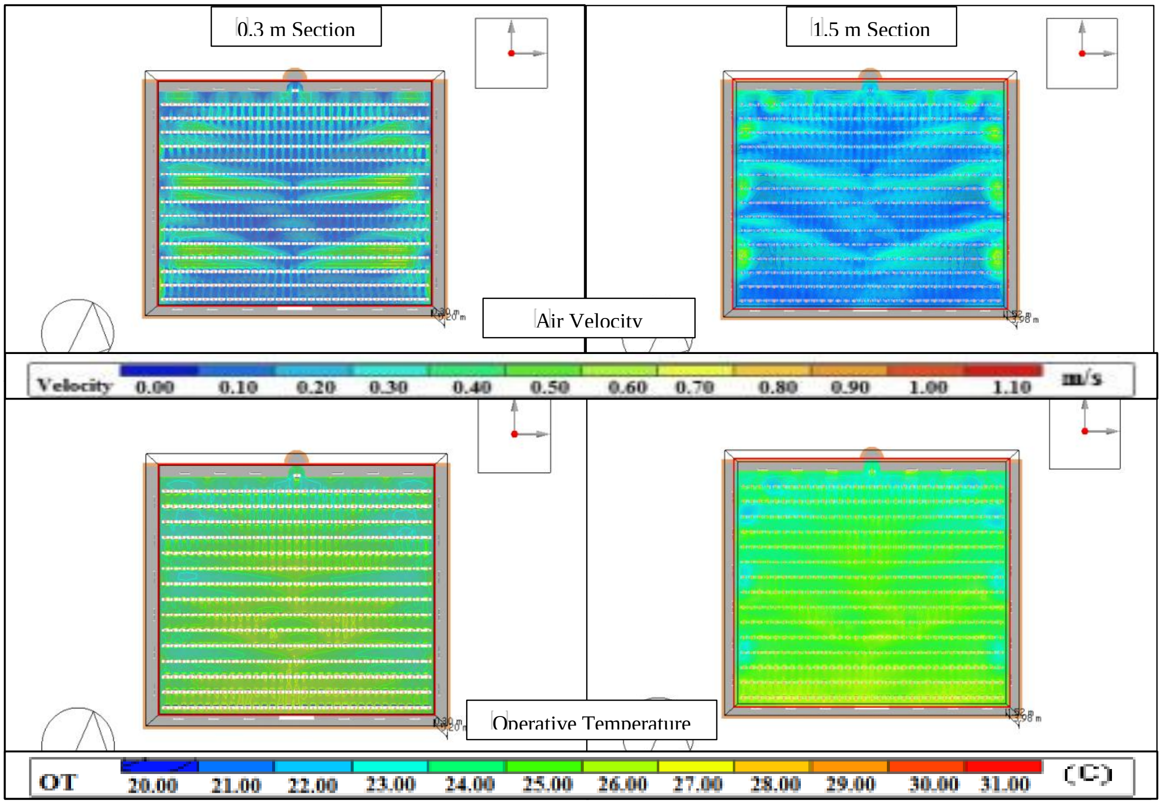


Figure 5.15: M3- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.15 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.16 for a diffuser discharge velocity of 3.5m/s. Air velocity contours did improve a little, varying between 0.1m/s to 0.4m/s in both sections with velocity above 0.3m/s usually occurring exactly below the diffuser location but in much reduced area, indicating that there would be temperature offset 1^{o}C or less. Operative Temperature was varying between 23^{o}C to 26^{o}C with 25^{o}C prevalent in the whole space. Thus PMV was seen falling between -1.0 at the perimeter region to -0.5 to 0.0 in the middle region. The resultant PPD for both sections was found to vary from 5% to 30%

with higher value occurring in the region below diffusers demonstrating that space can be termed thermally comfortable for 80% or higher occupants.

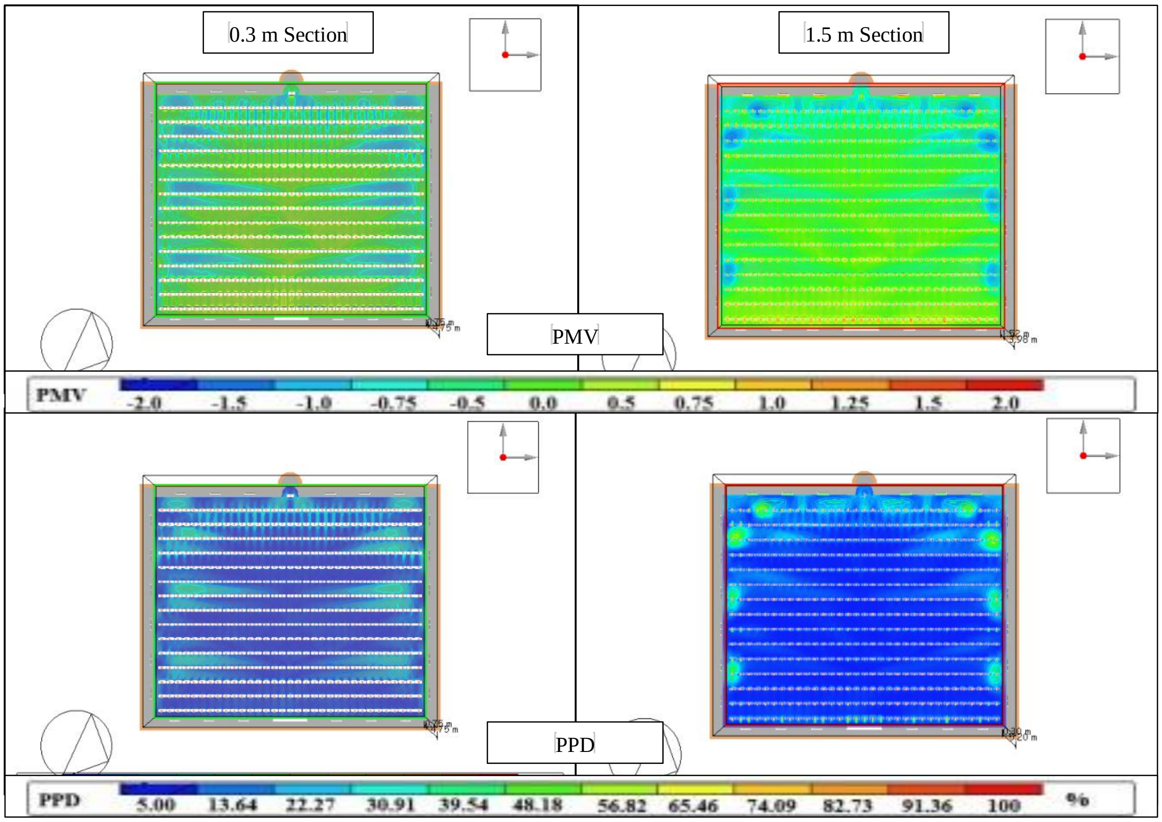


Figure 5.16: M3- PMV and PPD contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

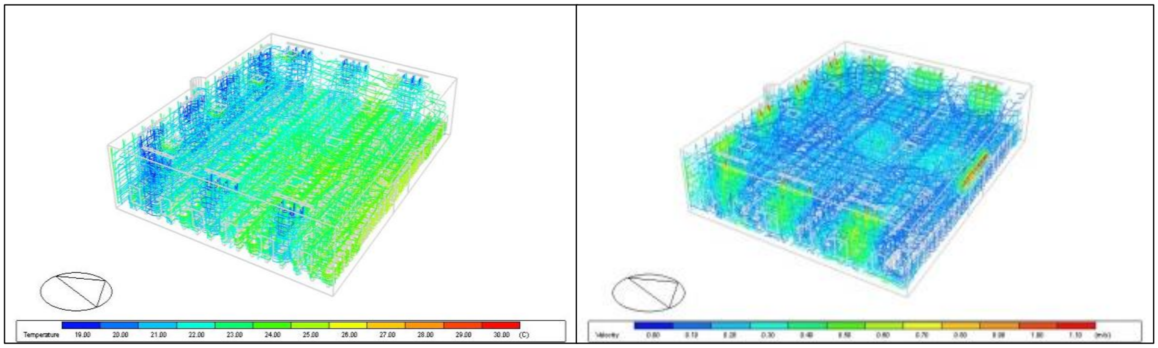


Figure 5.17: M3- 3-D Temperature and Velocity Contours for 3.5 m/s diffuser discharge velocity

From this analysis, it was observed that change in return diffuser location had a significant impact on improving thermal comfort and higher diffuser discharge value reduced the draft regions by striking a balance between buoyancy and convection effect. But the uniform temperature value of 24^{o}C and uniform velocity value of 0.13 m/s that was assumed in the EnergyPlus was not achieved. Figure 5.17 displays 3-D contours of Temperature and Velocity that were obtained for 3.5 m/s diffuser discharge velocity case where average temperature observed was 23^{o}C and average velocity observed was 0.3 m/s.

5.2.2 Through-Wall Air Distribution (TWAD) (M4 and M5)

5.2.2.1 TWAD with linear/slot supply diffusers and return on Ceiling (M4)

M5 air distribution scheme was simulated for thermal comfort performance. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.18 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.19 for a diffuser discharge velocity of 1.5m/s. Air velocity was found to be varying between 0.1m/s to 0.6m/s in 0.3 m section with velocity above 0.5m/s being predominant. In 1.5m section it was found to be varying between 0.1m/s to 0.4m/s with 0.2m/s predominant. This means that there will be temperature offset occurring in almost all locations upto a value of 4^{o}C as the difference between air temperature and MRT in most locations is around 6^{o}C resulting in large cool area in most of the occupied zone. Operative Temperature also resulted towards cold region, varying between 21^{o}C to 24^{o}C in 0.03m section to 23- 24^{o}C in 1.5m section mainly influenced by air temperature. There was temperature stratification observed in the occupied zone from 0.3 m section to 1.5 m section. PMV was found to vary from -2, which is not desirable in any situation predominantly occurring in 0.3m section, to -0.50 mostly in 1.5m section. The PPD results for both sections showed a 100% dissatisfaction area which is not desirable in any situation. For this diffuser discharge velocity the space can be termed as thermally not comfortable and a case of overcooling. These high variations in the two sections were due to the low diffusers discharge velocity which resulted in small throw distance. The diffuser discharge velocity was increased to 2 m/s and results showed no improvement in the thermal comfort status. All most all the parameters showed similar results as compared to previous velocity case and only difference was the increase in the throw distance which resulted in cool spots moving towards center.

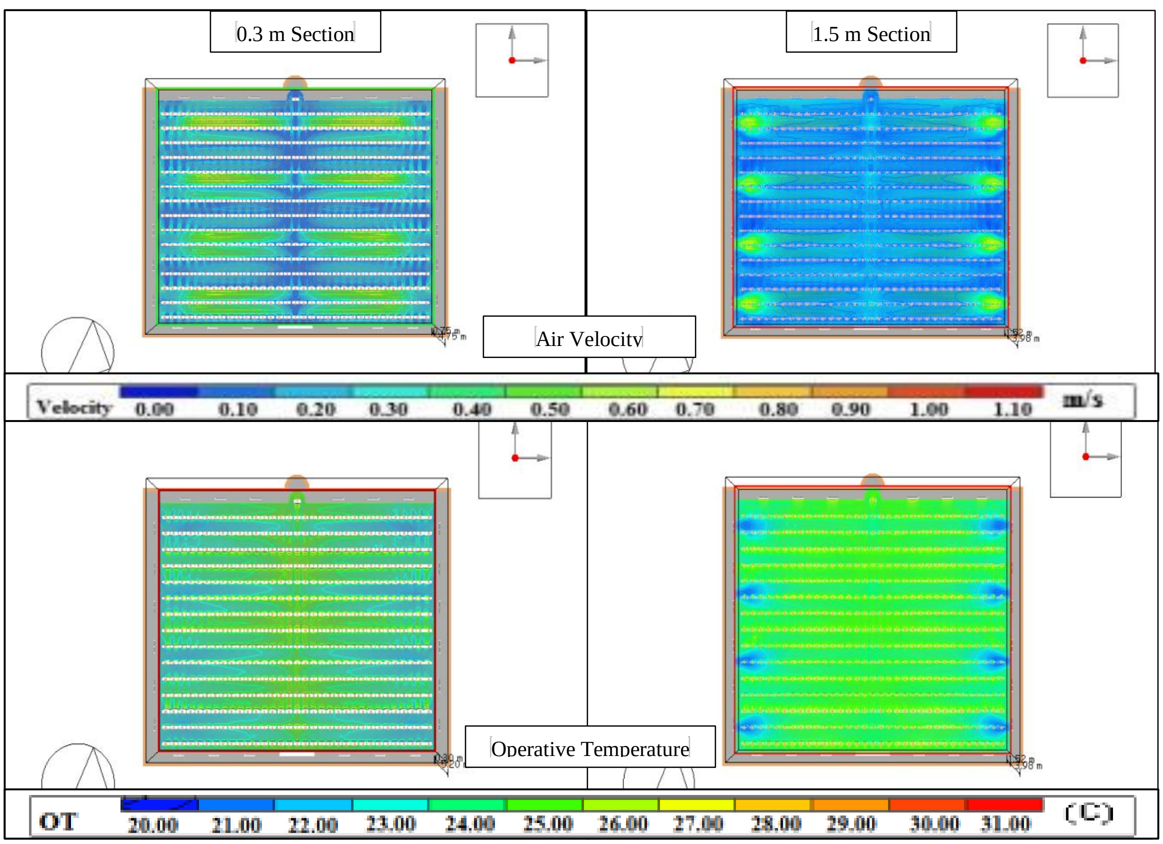


Figure 5.18: M4- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

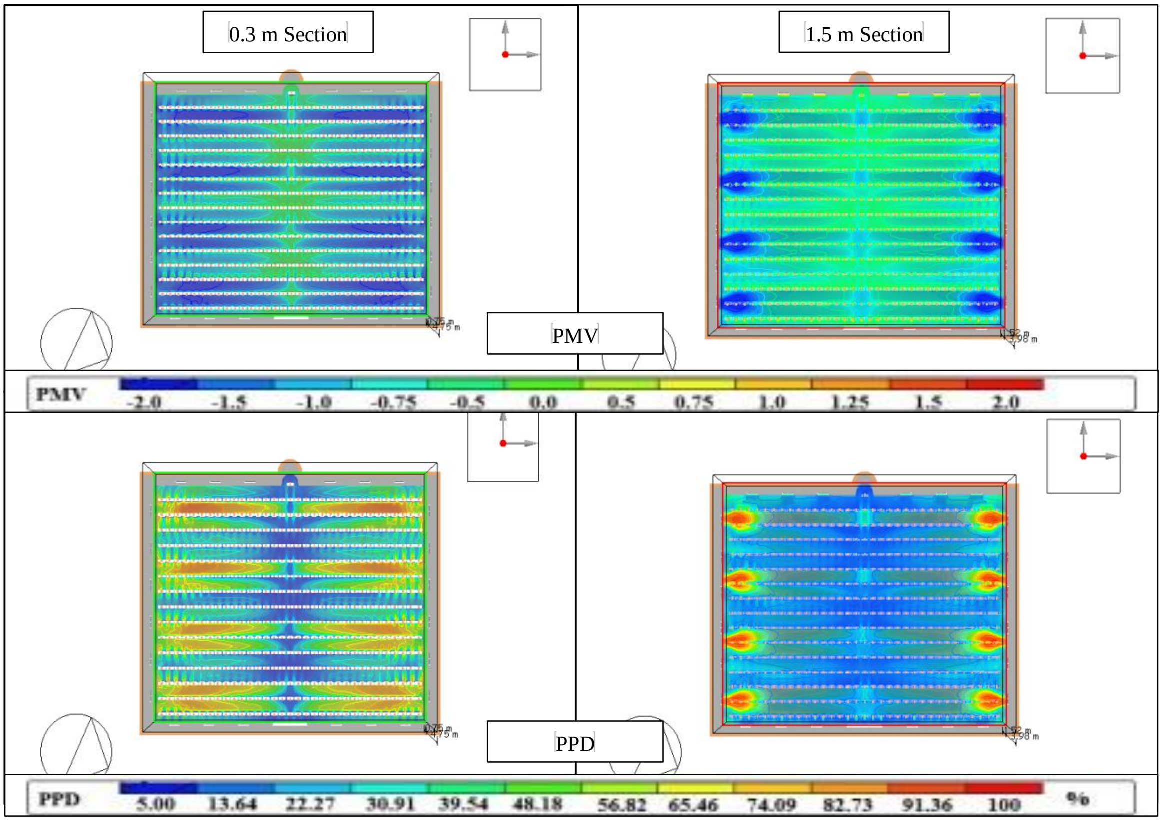


Figure 5.19: M4- PMV and PPD contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

Then the diffuser discharge velocity was changed to 3 m/s. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.20 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.21 for a diffuser discharge velocity of 3 m/s. Air temperature was found to vary between 19ºC to 23ºC in both sections with 21ºC being predominant which is towards cold region. Air velocity contours were found to be varying between 0.1m/s to 0.6 m/s with velocity above 0.6 m/s occurring at the throw areas causing temperature offset in most locations upto a value of 4ºC. Operative Temperature also resulted towards cold region, varying between 22ºC to 24ºC in 0.03m section to 23- 24ºC in 1.5m section mainly influenced by air temperature. PMV was found to vary from -2 to -0.75. The PPD results for both sections showed a 90% dissatisfaction areas. The diffuser discharge velocity was further increased to 3.5 m/s but there was no visible improvement. From these results it can be said that using a single set point temperature to design HVAC system irrespective of the air distribution scheme will cost on thermal comfort status. With this air distribution scheme there is lot of potential for energy conservation without compromising on thermal comfort.

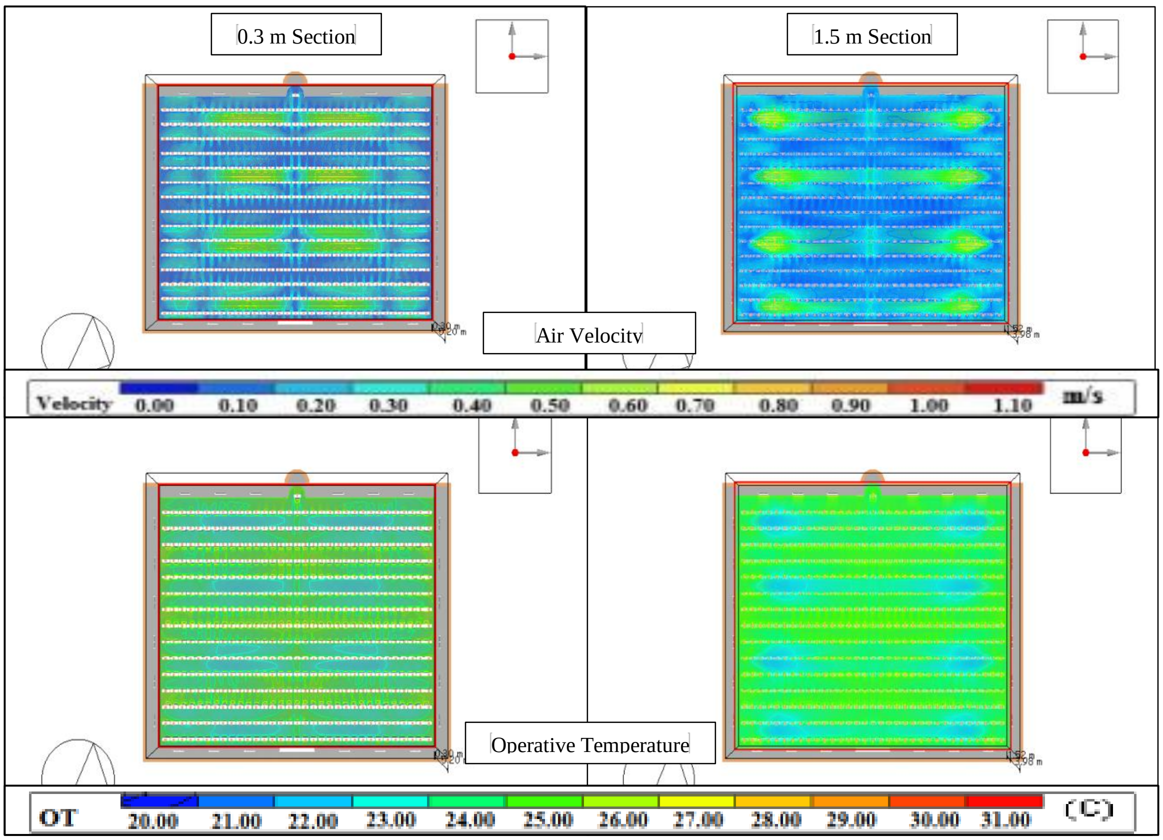


Figure 5.20: M4- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 3 m/s velocity

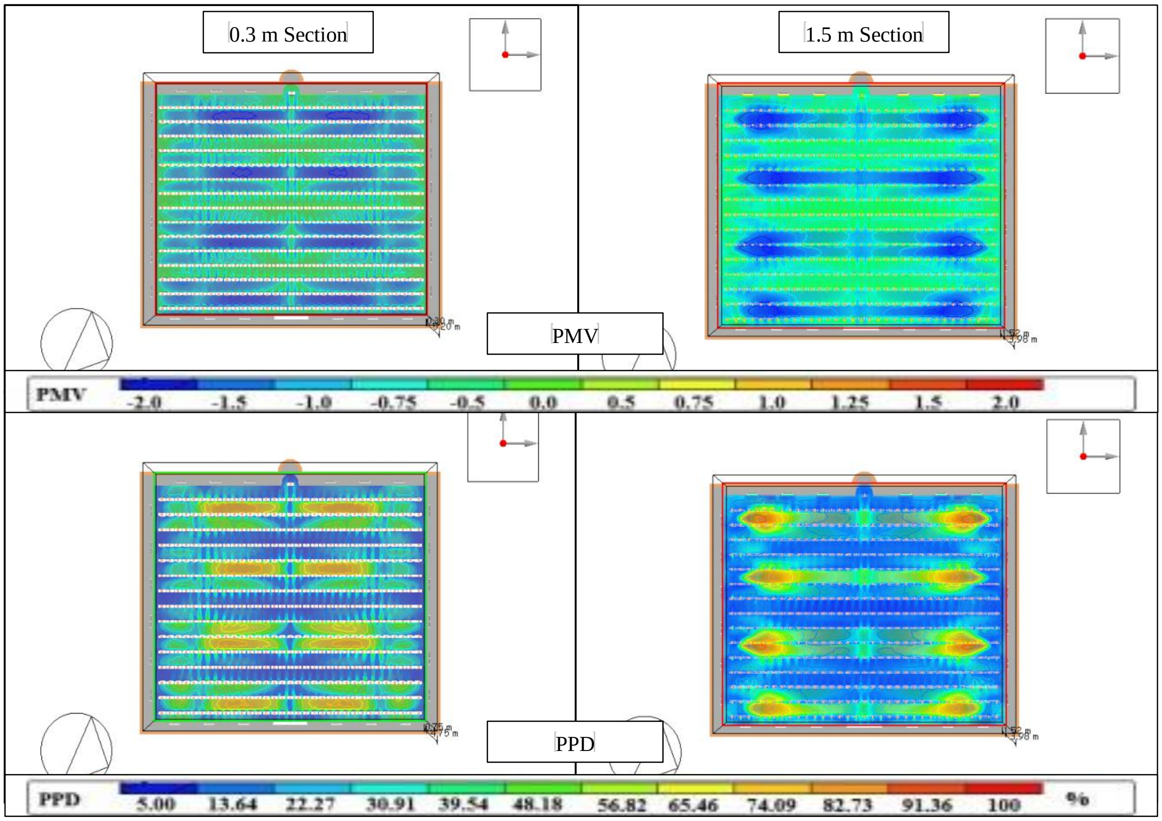


Figure 5.21: M4- PMV and PPD contours at sections 0.3 m and 1.5 m for 3 m/s velocity

5.2.2.2 TWAD with linear/slot supply diffusers and return on wall (M5)

M4 air distribution scheme was simulated for thermal comfort performance. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.22 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.23 for a diffuser discharge velocity of 1.5m/s. Air velocity contours were found to be varying between 0.1m/s to 0.5m/s with 0.5 m/s velocity occurring in the lower region of the occupied zone because of the buoyancy effect. In 1.5m section air velocity of 0.5 m/s magnitude was observed in the diffuser throw area. There will be temperature offset occurring in regions of high velocity upto a value of 4^{o}C since MRT and air Temperature difference is around 7^{o}C. Operative Temperature resulted towards cool value, varying between 22^{o}C to 24^{o}C in 0.3m section to 21- 24^{o}C in 1.5m section mainly influenced by air temperature with 23^{o}C being predominant. PMV was found to vary from -2, which is not desirable in any situation predominantly occurring in 0.3m section, to -0.50 mostly in 1.5m section. The PPD results for both sections showed a 100% dissatisfaction area which is not desirable in any situation. Results of this model compared to M4 model for same diffuser discharge velocity are better because of the extra diffusers which reduced the volume flow rate in the diffusers.

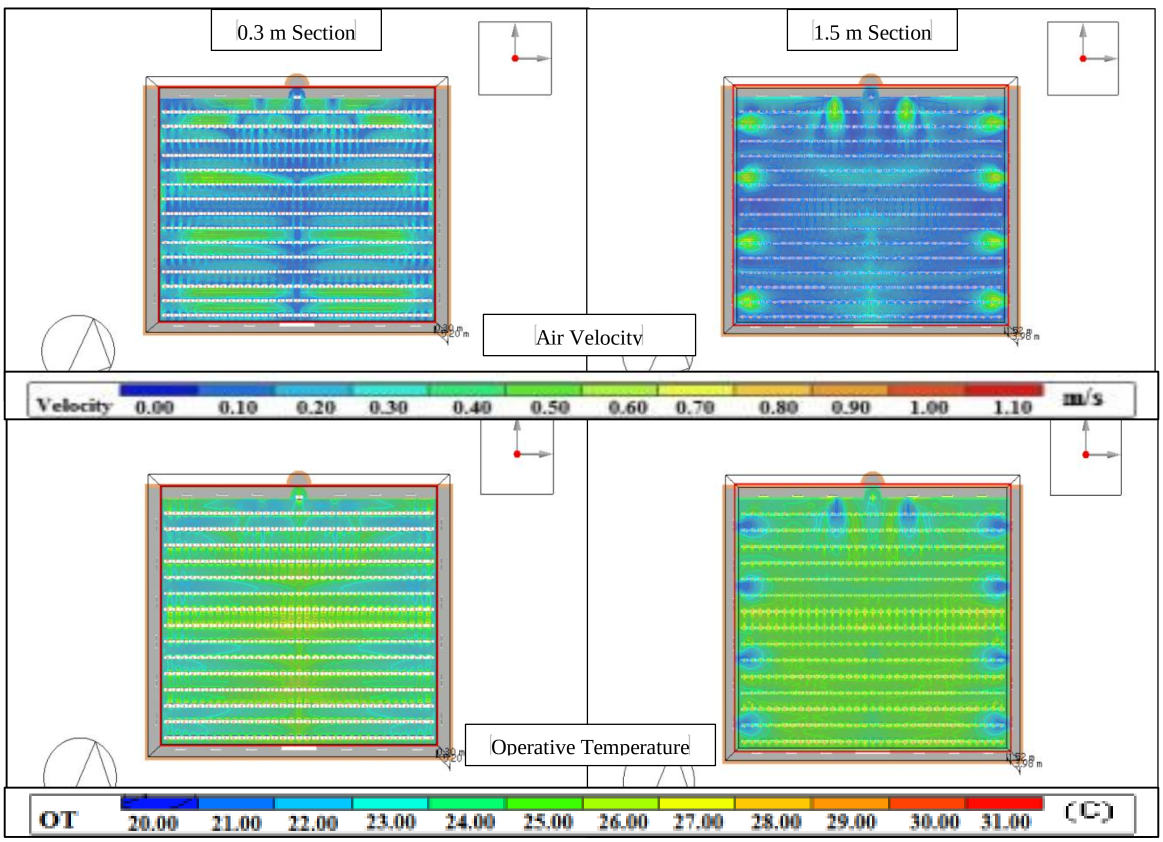


Figure 5.22: M5- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

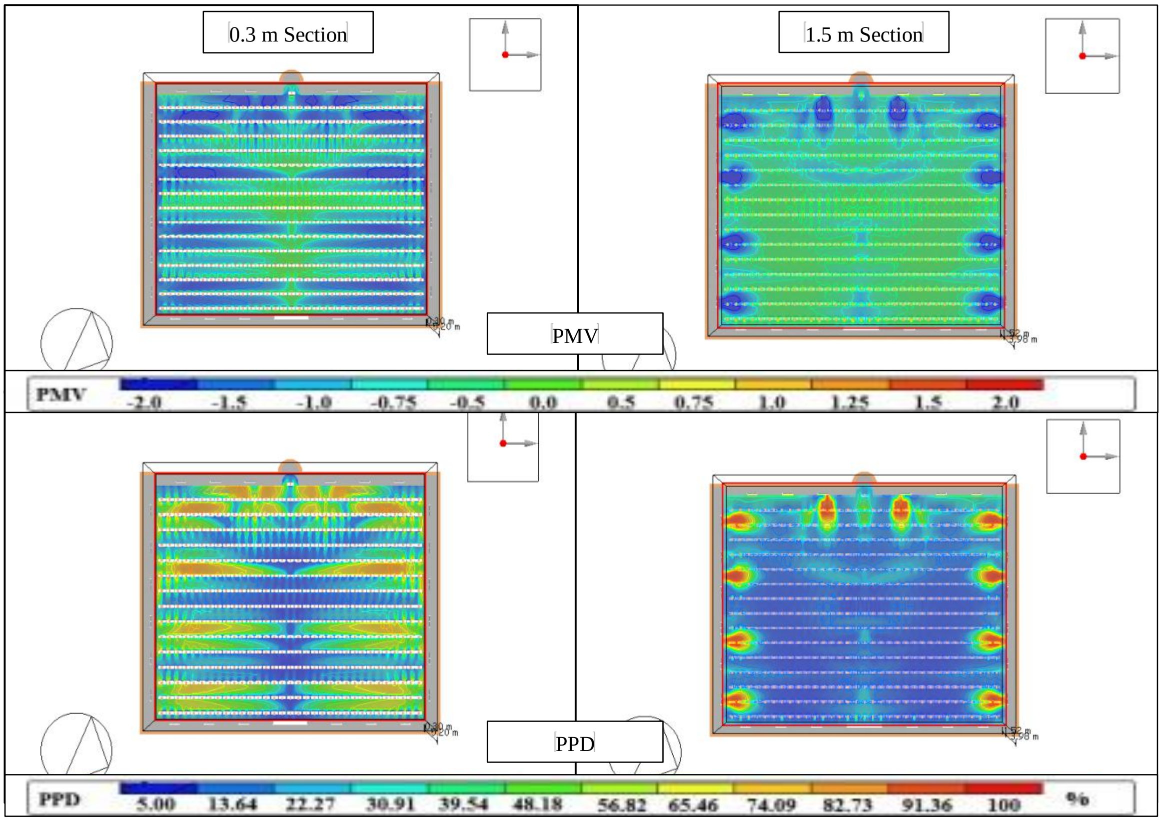


Figure 5.23: M5- PMV and PPD contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

The diffuser discharge velocity was increased to 2 m/s which gave similar results compared to previous case and only increased the throw distance of the diffuser. Velocity 3 m/s was used as diffuser discharge velocity. Air velocity contours were found to be varying between 0.1 m/s to 0.5 m/s in both sections and the throw distance was further increased. Velocity at the throw area was highest indicating temperature offset upto a value of 4°C at those locations. Similar to previous case, operative temperature was found to vary between 22°C to 24°C in both sections with 22°C being predominant. PMV was found to vary from -2, which is not at all desirable in any situation predominantly occurring in 0.3m section, to -0.5 mostly in 1.5m section. This resulted in PPD of 100% in the location of the drafts.

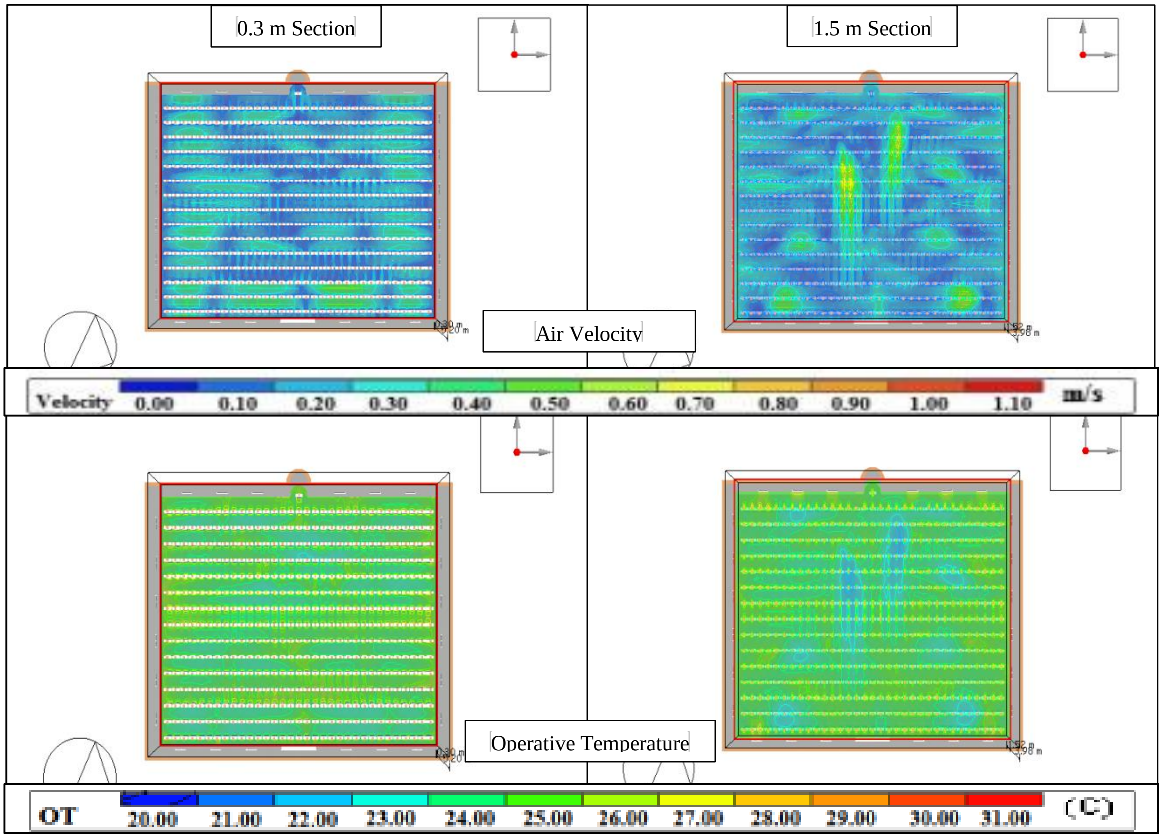


Figure 5.24: M5- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

Lastly a diffuser discharge velocity of 3.5 m/s was tested. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.24 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.25 for a diffuser discharge velocity of 3.5 m/s. Air temperature for this velocity were found to vary between 21°C to 24°C in both sections with 22°C being predominant which is towards cold region. Air velocity contours were found to be varying between 0.1m/s to 0.5 m/s in 0.3m section with very spot of velocity above 0.3 m/s which show the effect of increasing the discharge velocity. Although there was occurrence of temperature offset in most locations upto a value of 3°C in both sections. PMV was found to vary from -1.5, which is not at all desirable in any situation predominantly occurring in 1.5m section, to -1 to -0.5 mostly in 0.3m section. This resulted in PPD of 100% in the location of the drafts, with this velocity adding more throw distance. One important thing noticed here is that this value of velocity caused the air to mix more uniformly, but it shifted the space to being over cooled.

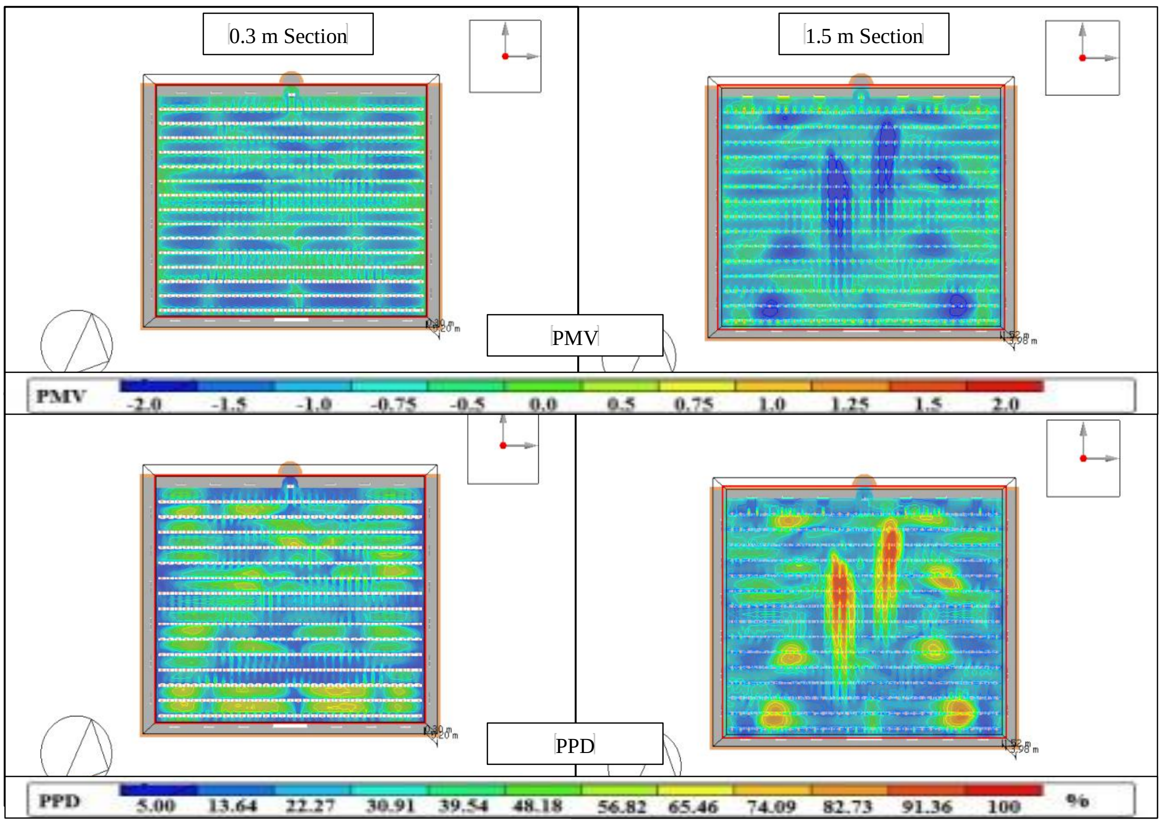


Figure 5.25: M5- PMV and PPD contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

5.2.3: Under-Floor Air Distribution (UFAD) (M6 and M7)

5.2.3.1 UFAD with linear/slot supply diffusers and return on Ceiling (M6)

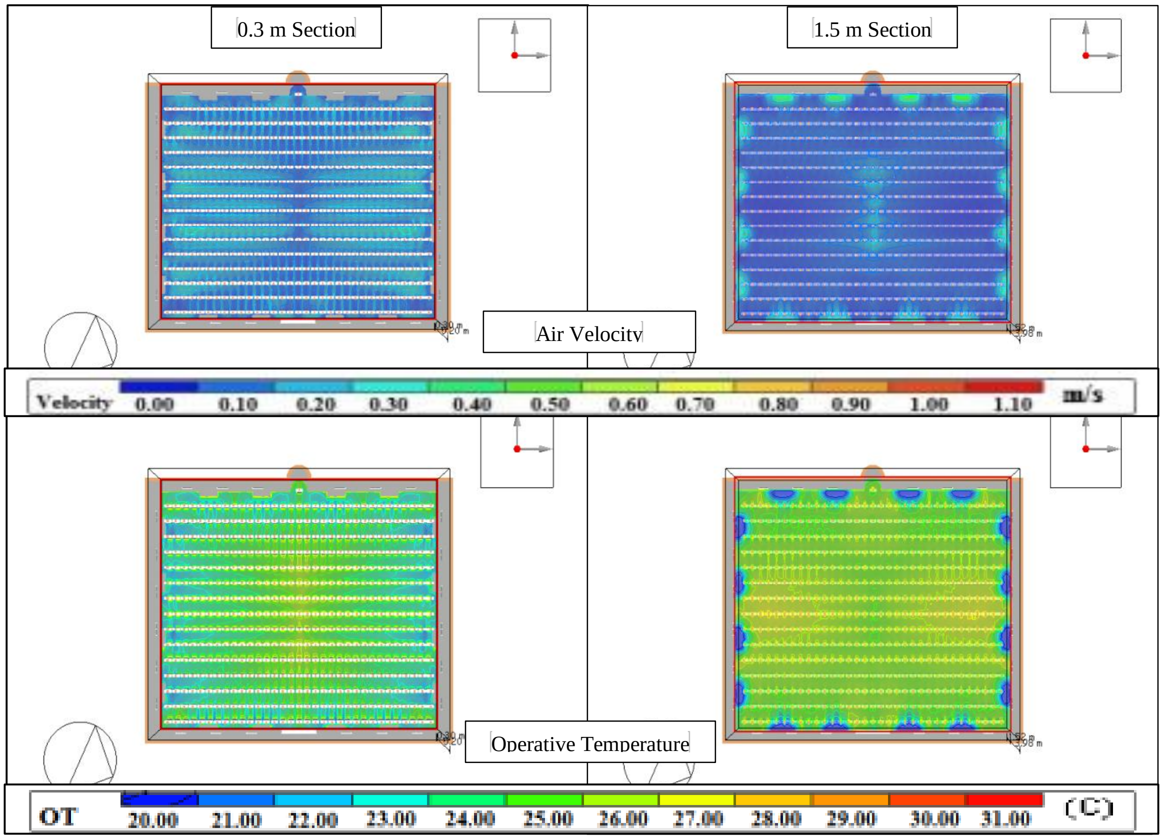


Figure 5.26: M6- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 0.8 m/s velocity

M6 air distribution scheme was simulated for thermal comfort performance. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.26 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.27 for a diffuser discharge velocity of 0.8m/s. In the occupied zone, air temperature stratification was seen to exceed the allowable limit of 3^{o}C. Air velocity was found to varying between 0.1m/s to 0.2m/s in both sections which conforms with the comfort limit and does not allow any temperature offset. Operative Temperature resulted towards cold region, varying between 21°C to 25°C in 0.03m section to 22- 26°C in 1.5m section, and a value of 20°C mainly occurring near diffusers. PMV was found to vary from -2, predominantly occurring in 0.3m section, to -0.7 mostly observed in 1.5m section. This variation among sections is due to temperature stratification. This resulted in PPD of 100% in the location of the drafts. For this diffuser discharge velocity the space can be termed as thermally not comfortable and occupants will feel **"cold"**. The discharge velocity was increased to 1 m/s and then to 1.25 m/s with almost no change in the parameter contours.

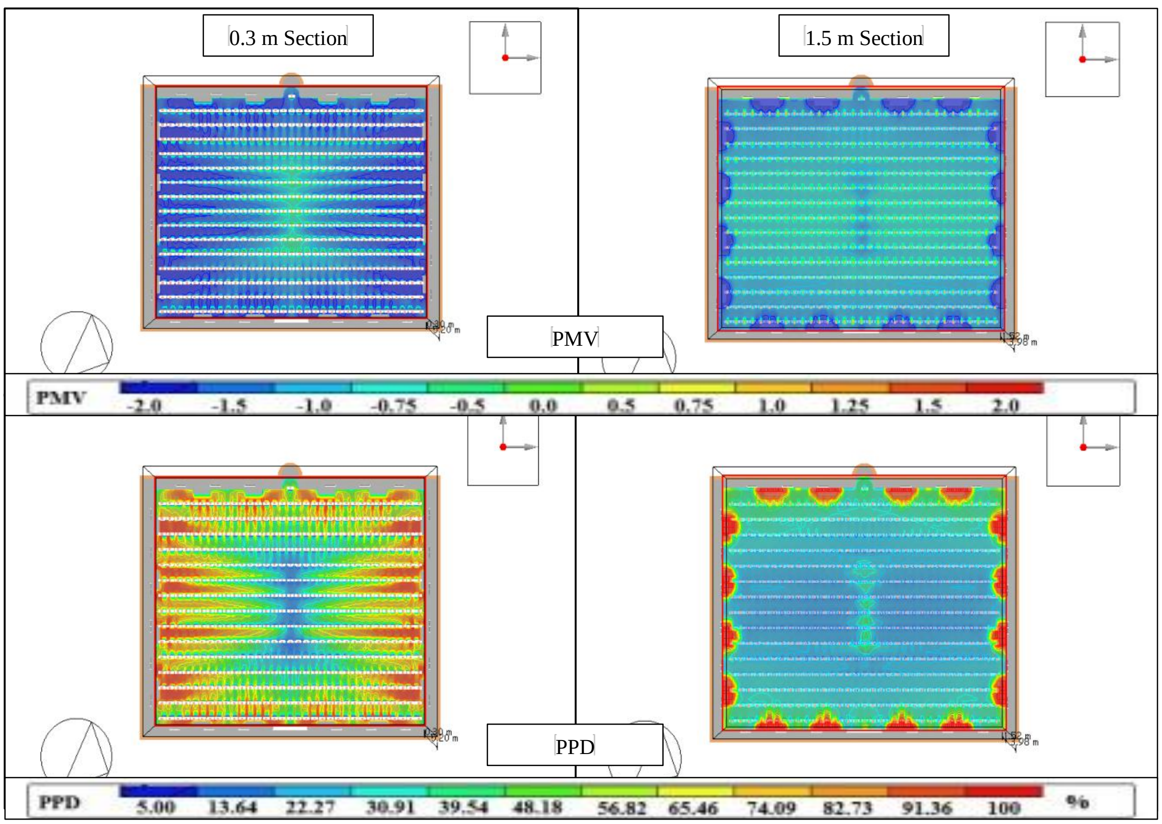


Figure 5.27: M6- PMV and PPD contours at sections 0.3 m and 1.5 m for 0.8 m/s velocity

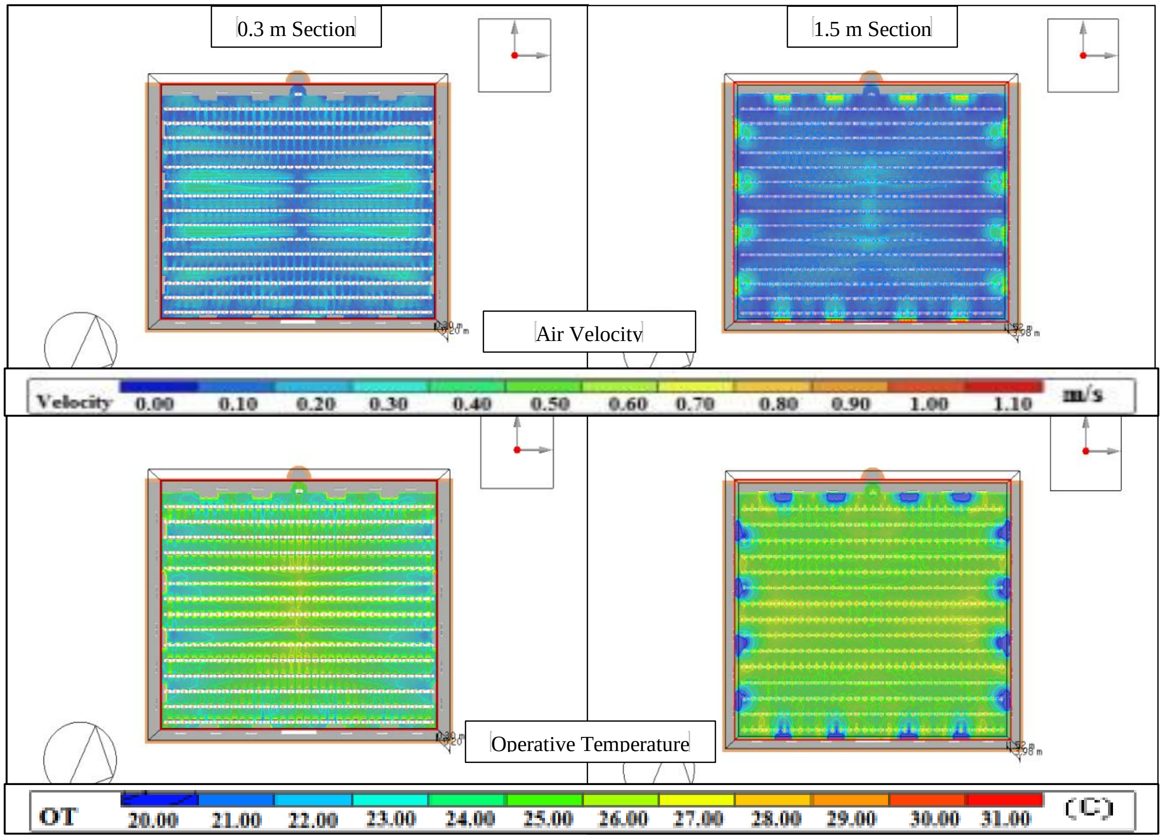


Figure 5.28: M6- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

The diffuser discharge velocity was further increased to 1.5 m/s. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.28 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.29 for a diffuser discharge velocity of 1.5 m/s. Again no significant change was observed in any of the parameters except for air velocity compared to previous case. Air velocity was found to vary between 0.1m/s to 0.4m/s in 0.3m section with increased 0.3m/s velocity areas and in 1.5m section, it was found to vary between 0.1m/s to 0.7m/s with the latter occurring near the diffusers. Operative Temperature also resulted towards cold region, varying between 22^{o}C to 25^{o}C in 0.3m section to 20- 26^{o}C in 1.5m section, 20^{o}C mainly occurring near diffusers. PMV was found to vary from -2, predominantly occurring in 0.3m section, to -0.7 mostly observed in 1.5m section. Thus PPD of 100% was prevalent in 0.3m section while in 1.5m section it occurred near diffusers and in 1.5m section it was 30% PPD which dominated. This goes to show that UFAD is not effective in these circumstances.

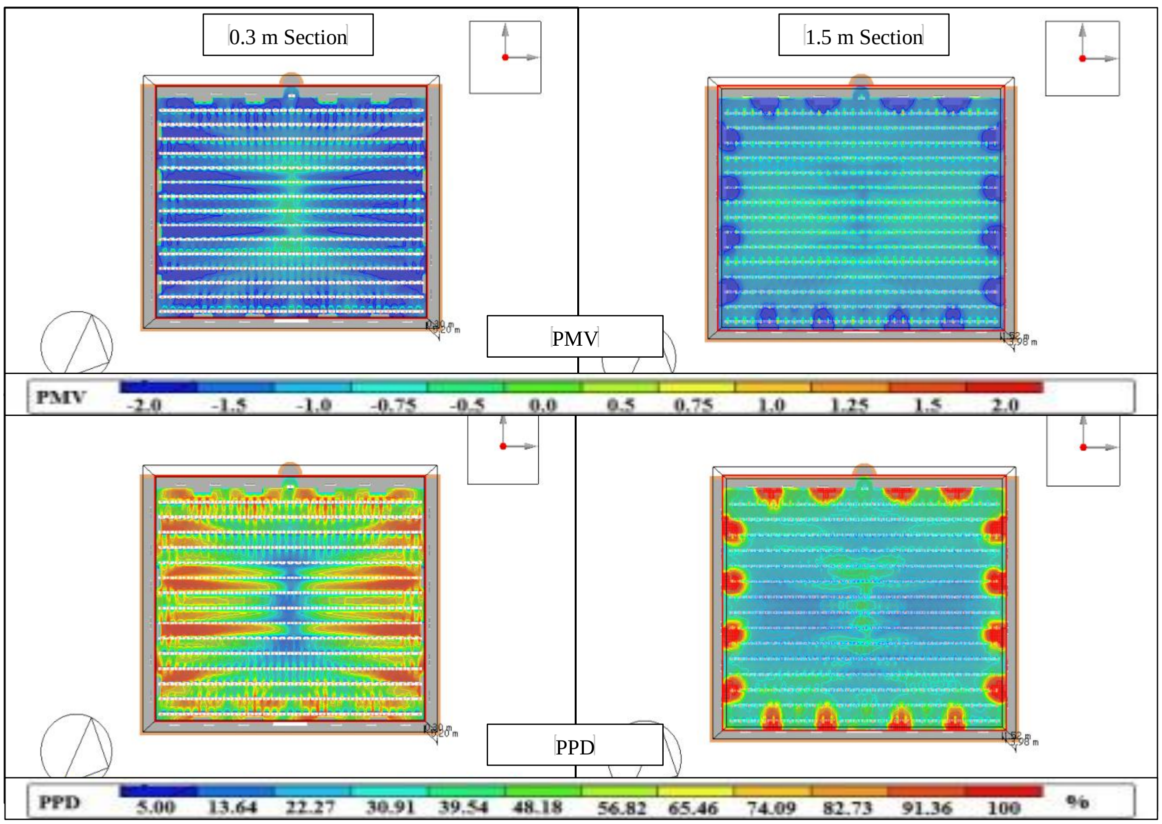


Figure 5.29: M6- PMV and PPD contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

5.2.3.2 UFAD with linear/slot supply diffusers and return on Wall (M7)

M7 air distribution scheme was simulated for thermal comfort performance. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.30 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.31 for a diffuser discharge velocity of 0.8m/s.

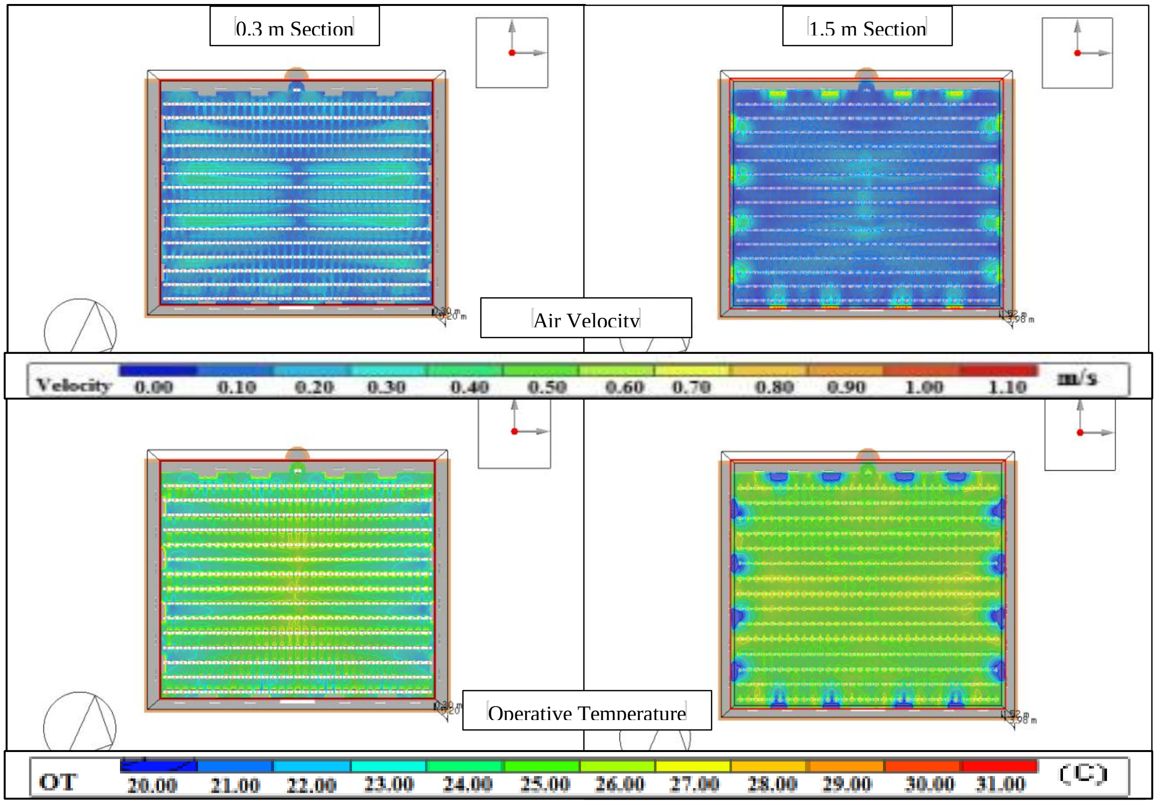


Figure 5.30: M7- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 0.8 m/s velocity

The contours are similar to M6 model contours at same velocity. Air velocity contours were found to be varying between 0.1m/s to 0.2 m/s in both section which conforms with the comfort limit and does not allow any temperature offset. Operative Temperature also resulted towards cold region, varying between 22^{o}C to 25^{o}C in 0.3m section to 20- 26^{o}C in 1.5m section, 20^{o}C mainly occurring near diffusers. In the occupied zone, air temperature stratification was seen to exceed allowable limit of 3^{o}C. PMV was found to vary from -2, predominantly occurring in 0.3m section, to -0.7 mostly observed in 1.5m section. This resulted in PPD of 100% in the location of the drafts. For this diffuser discharge velocity the space can be termed as thermally not comfortable and occupants will feel cold.

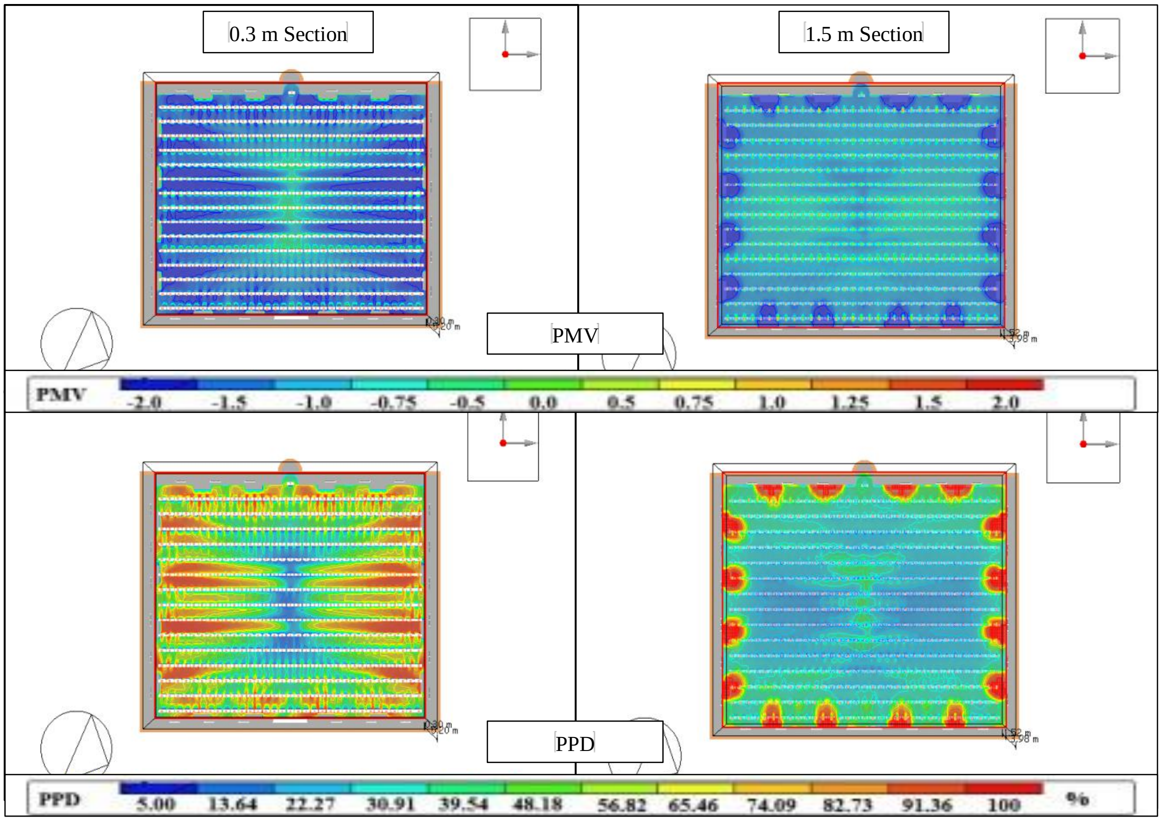


Figure 5.31: M7- PMV and PPD contours at sections 0.3 m and 1.5 m for 0.8 m/s velocity

The diffuser discharge velocity was further increased to 1 m/s, 1.25 m/s and then to 1.5 m/s with no significant change compared to previous model at these velocities. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.32 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.33 for a diffuser discharge velocity of 1.5m/s.

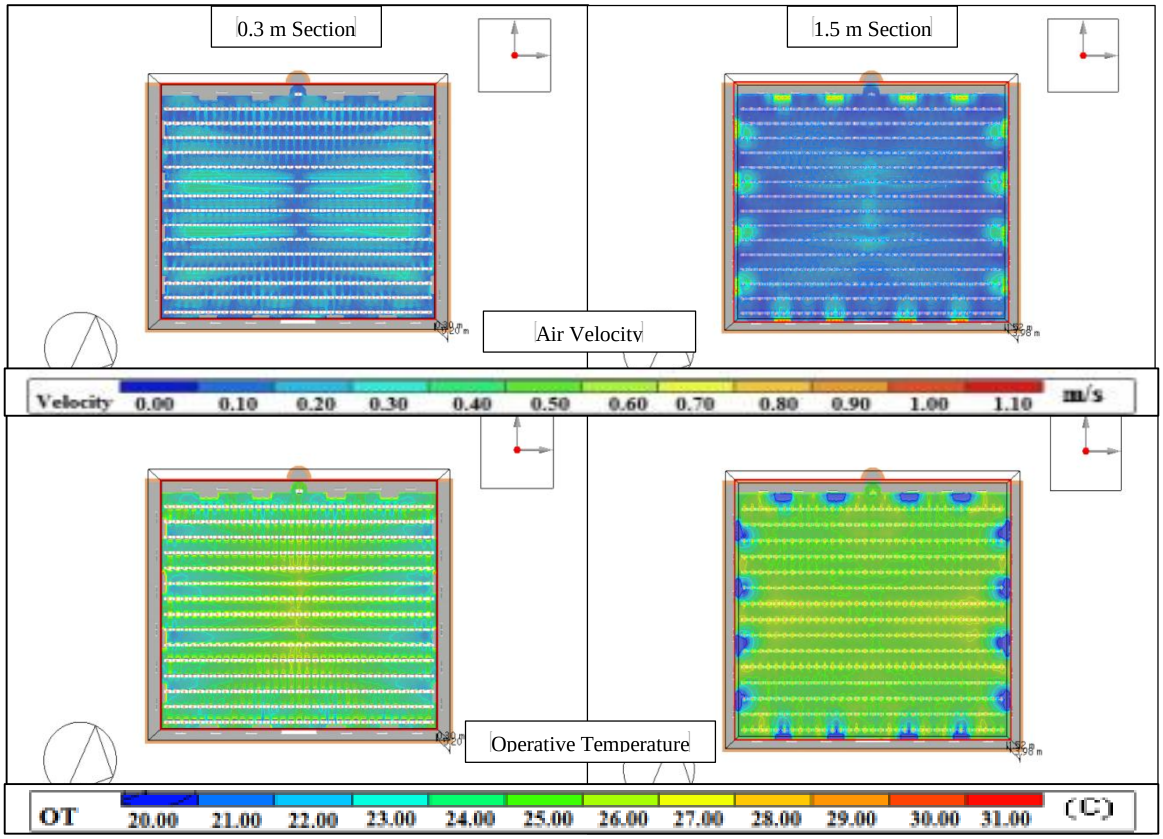


Figure 5.32: M7- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

Air velocity contours were found to be varying between 0.1m/s to 0.2 m/s in both section which conforms with the comfort limit and does not allow any temperature offset. Operative Temperature also resulted towards cold region, varying between 22^{o}C to 25^{o}C in 0.3m section to 20- 26^{o}C in 1.5m section, 20^{o}C mainly occurring near diffusers. In the occupied zone, air temperature stratification was seen to exceed allowable limit of 3^{o}C. PMV was found to vary from -2, predominantly occurring in 0.3m section, to -0.7 mostly observed in 1.5m section. This resulted in PPD of 100% in the location of the drafts. Since the thermal load in this study is uniformly distributed and the supply air temperature constrained by the assumptions of this study, the possible solution to enhance the performance of this strategy could not be achieved. Temperature stratification in these configurations was on the higher side due to predominant buoyancy effect.

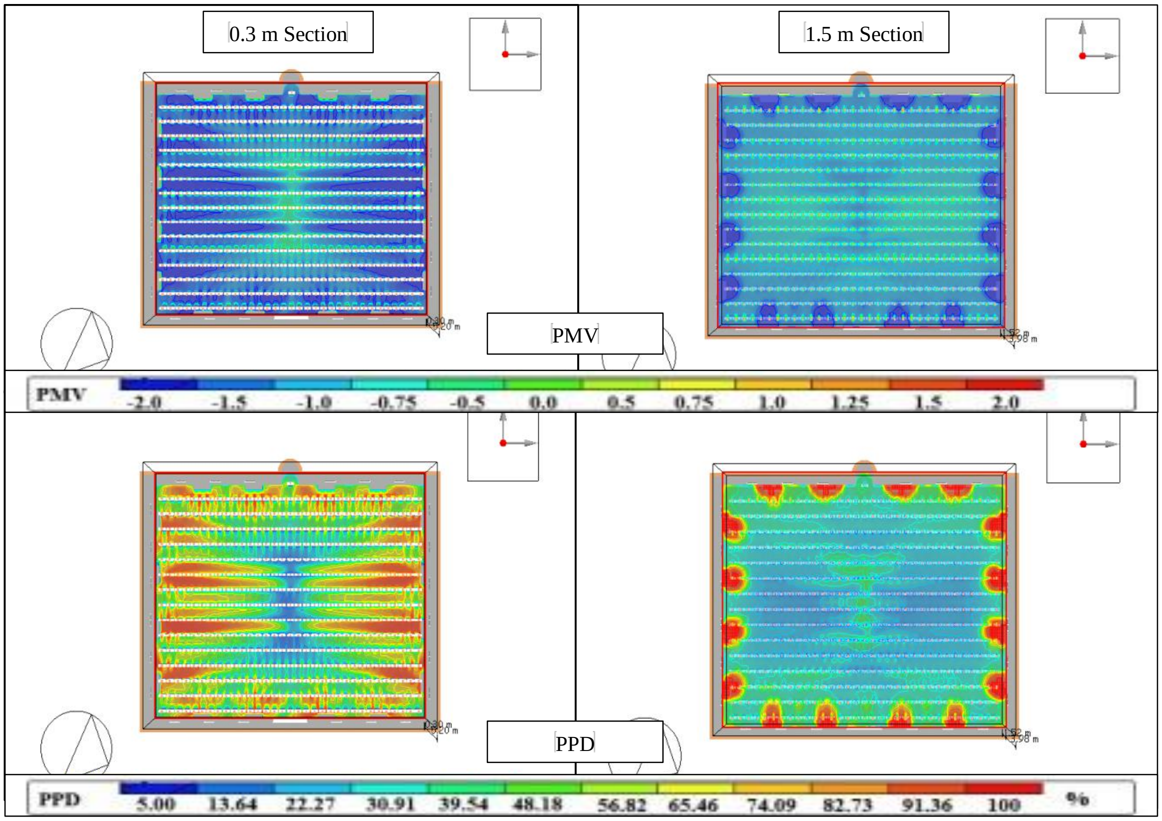


Figure 5.33: M7- PMV and PPD contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

5.3 Results Highlights of Continuous HVAC Operation

The objective of this part was to test the performance of base case model for energy and thermal comfort using energy simulation and CFD simulations. Energy simulation showed that thermal comfort performance of the base case model was within the specified PMV limit of 0.0 to 0.5. But the main assumption of the energy simulation software that the air mix to a uniform temperature of 24°C and velocity will be constant at 0.13m/s throughout the environment was not practical. In order to support this argument a detailed CFD analysis of commonly used air distribution strategies was conducted. There were 7 models created for this purpose that included 3 CBAD models, 2 TWAD models and 2 UFAD models. Table 11 summarizes the results of this part of research study.

Table 5.1: Summary for thermal comfort results with continuous HVAC operation

Model No.	Parameters	Velocity Cases				Results
		Case 1	Case 2	Case 3	Case 4	
M1	Temperature (C)	21-24	22-24	23-25	23-25	Cool spots were observed.
	Velocity (m/s)	0.1-0.5	0.1-0.5	0.1-0.5	0.1-0.5	
M2	Temperature (C)	21-24	21-24	22-24	22-24	Cool spots were observed.
	Velocity (m/s)	0.1-0.5	0.1-0.5	0.1-0.5	0.1-0.5	
M3	Temperature (C)	21-24	21-24	22-24	22-24	Achieved Thermal Comfort requirements
	Velocity (m/s)	0.1-0.5	0.1-0.5	0.1-0.4	0.1-0.4	
M4	Temperature (C)	19-23	19-23	19-23	19-23	Cold spots were observed.
	Velocity (m/s)	0.1-0.5	0.1-0.5	0.1-0.6	0.1-0.7	
M5	Temperature (C)	19-23	19-23	19-23	19-23	Cold spots were observed.
	Velocity (m/s)	0.1-0.5	0.1-0.5	0.1-0.6	0.1-0.6	
M6	Temperature (C)	19-22	19-22	19-22	19-22	Cold spots were observed.
	Velocity (m/s)	0.1-0.2	0.1-0.2	0.1-0.4	0.1-0.4	
M7	Temperature (C)	19-22	19-22	19-22	19-22	Cold spots were observed.
	Velocity (m/s)	0.1-0.2	0.1-0.2	0.1-0.4	0.1-0.4	

- CBAD air distribution strategies performed best even though average air temperatures were between 21°C to 23°C with M3 model giving best results by maintaining maximum occupied zone at 5% PPD. Although the average air temperature in the occupied zone was around 23°C which is less than the set point temperature, and average air velocity around 0.25m/s in the occupied zone which is higher than the software assumption of 0.13m/s, overall thermal comfort conditions were within PMV of -0.5 and 0.5. The article by Al-Ajmi [6] in which author presented a study on thermal comfort performance of mosques, the air temperature and air velocity reported were similar to the results of CBAD strategy. This strategy balanced the effect of buoyancy and convection and benefited from it to maximum level.
- TWAD strategy was towards cold side because supply air was very close to the occupied zone. The average air velocity in these strategies was varying between 0.2m/s and 0.5m/s which resulted in velocity drafts which caused offset of air temperature by a value greater than 3°C, while average air temperature observed in this strategy varied between 21°C to 23°C similar to what observed in CBAD. It uses the convection effect to good condition but this strategy would perform better when the loads are envelope dominate or in other words for higher values of MRT.
- UFAD strategy was the worst of the three with average air temperature 20°C and below although average air velocity was 0.3m/s and below. Stratification in occupied zone exceeded the limit of 3°C. This performance may be due to low supply air temperature of 12°C while the literature stressed use of 15°C to 18°C as the supply temperature with high number of outlets possible[10]. These temperatures were out of the scope of this research work.

5.4 Energy and Thermal comfort Analysis of Intermittent Strategy

5.4.1 Energy Performance

The base case model had continuous HVAC operation which means that HVAC system was working 24x7 even though occupancy occurred for 5 intermittent periods. Thus intermittent HVAC operation strategy was opted which will start the HVAC system 1 hour before start of the occupancy and shut it off at the end of the occupancy. In this way HVAC system was operating for 9 and 1/2 hours with 2 hours each during Fajr, Dhuhr and Asr prayers and 3 and half hours continuous during Maghreb and Isha Prayers, instead of 24 hours. Mosque building with new HVAC operation strategy was simulated using state-of-the-art DesignBuilder simulation for annual simulation using the weather data file of Dhahran for the year 2012. Results of total annual energy consumption for each month for continuous and intermittent operation are shown in Figure 5.34 and total cooling energy consumption for each month for continuous and intermittent operation are shown in Figure 5.35. A total of 85218.5 kWh or 181.62 kWh/m^2 of energy was consumed by the base case model annually of which 157.80 kWh/m^2 going for cooling alone. With the new operation strategy the annual energy consumption was reduced to 59787.22 kWh or 127.42 kWh/m^2, which accounts for a saving of 30% annually. Total cooling energy was reduced to 48612.06 kWh or 103.60 kWh/m^2, which constituted a savings of 35% of cooling energy. The monthly high was again observed in august at 9497.28 or 20.24 kWh/m^2 and lowest in February at 1222.77 or 2.6 kWh/m^2. The maximum savings was observed in the month of July at 41% energy savings and lowest was in the month of March at 2.8%. Going by the numbers, energy savings are excellent but these are not acceptable if the thermal comfort of the occupants in compromised. Thus require a detailed thermal comfort analysis.

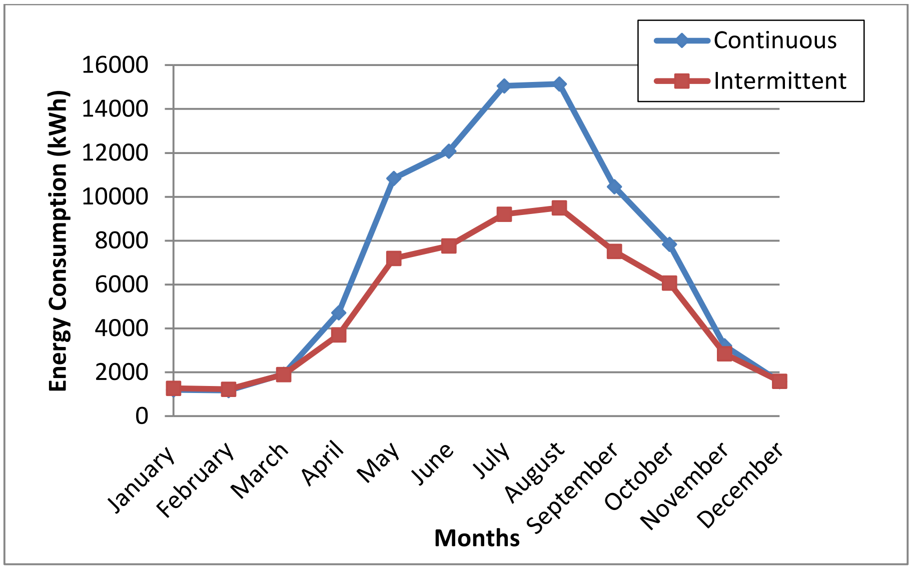


Figure 5.34: Total Annual Energy Consumption Comparison for Continuous and Intermittent Operations

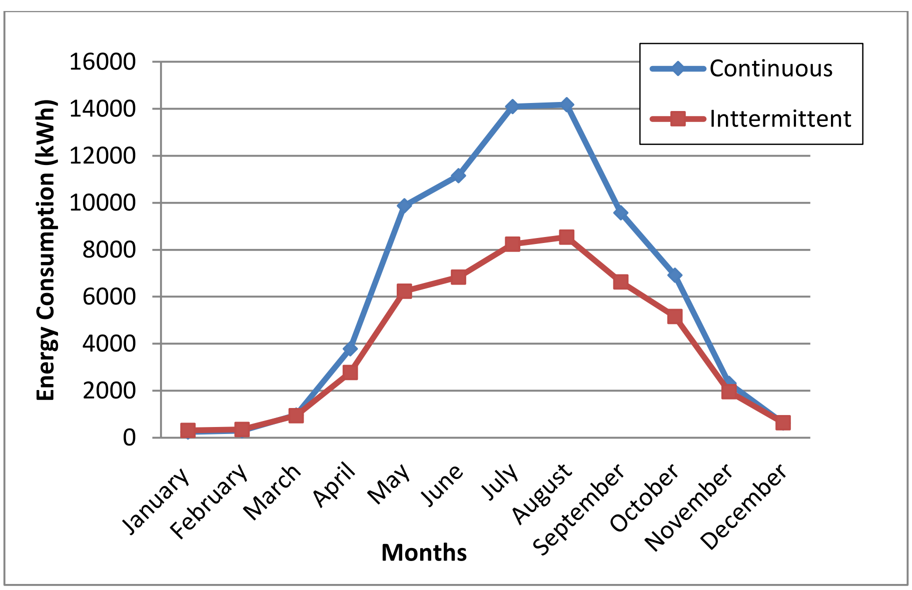


Figure 5.35: Total Cooling Energy Consumption Comparison for Continuous and Intermittent Operations

5.4.2 Thermal Comfort Performance

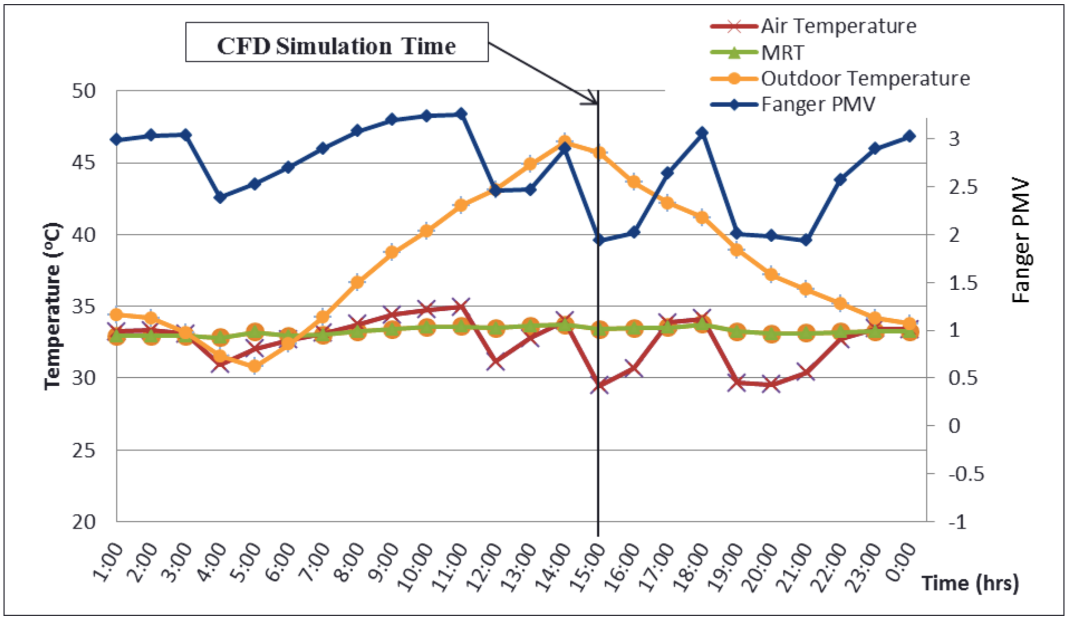


Figure 5.36: Thermal comfort analysis of Operation Strategy

Similar to base case thermal comfort analysis, simulation for comfort analysis of operation strategy was done on 21st July which present a typical summer design day and results are presented in Figure 5.36. Software results show that the PMV varies between 1.9 and 2.5 during occupancy periods which correspond to PPD of 70% or more which is way beyond the ASHRAE standard 55 limits. Again it should be noted that the software assumes a uniform velocity of 0.13 m/s and air temperature which is varying between 29°C and 31.5°C during occupancy periods in this case, to be uniform throughout the space. As observed in the base case thermal comfort analysis this uniformity of values is very difficult or impossible to achieve in large spaces as the velocity and temperature vary in the space depending upon diffuser location, discharge velocity of diffuser, load distribution, location of the return diffuser etc. MRT for this simulation varied between 32.5°C to 34°C throughout the day. The results of base case air distribution simulation showed that space was over cooled in few of the strategies. These strategies have the potential to solve the thermal comfort problem appearing in this operation strategy case. Thus a detailed thermal comfort analysis of operation strategies using air distribution strategy was required.

5.5 Assessment of Thermal Comfort with intermittent HVAC operation

In this part of the research work, first models created during analysis of base case model were used and then required sensitivity analysis was done to models to check for further enhancement in comfort status. The thermal comfort judgment criteria will be same as discussed in the base case analysis. In order to have comparable results with base case results, the time for CFD simulations was chosen to be same but the temperature boundary condition were obtained from energy simulation of the operation strategy. As the surface temperature boundary conditions and the setup remain same for all the CFD simulations the resultant MRT would again remain constant for all CFD simulations. Thus the MRT contours for 0.3 m and 1.5 m sections are presented in Figure 5.37. Mean Radiant Temperature (MRT) values were observed to be around 33^{o}C to 34^{o}C for both sections, except near Sun-facing west side windows (1.5m section) where MRT was around 36^{o}C which is reasonable. The average value was almost similar to the EnergyPlus prediction. In operations strategy thermal comfort analysis for all CFD simulations a humidity ratio of 36.14% was used that was obtained from energy simulation at stated time and date. And also the air temperature predicted by the EnergyPlus for the stated time of the was 29.58^{o}C which the

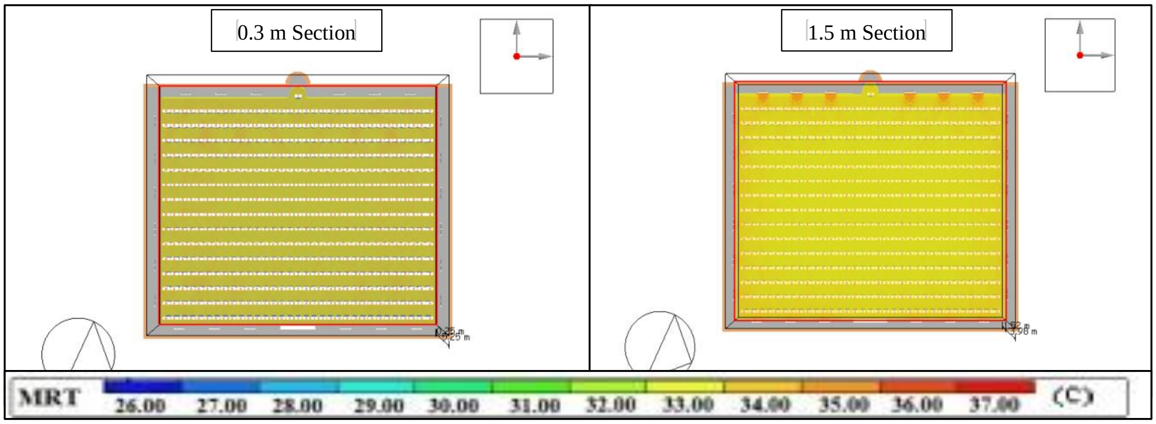


Figure 5.37: MRT contours for 0.3 m and 1.5 m sections for Operation Strategy

5.5.1 Ceiling-Based Air Distribution (CBAD) (M1, M2, M3 and M3-1)

The setup for these schemes was same as in base case simulations and an additional scheme M3-1 was used when compared to base case.

5.5.1.1 CBAD with four way supply diffusers (M1)

M1 air distribution scheme was simulated for thermal comfort performance. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.38 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.39 for a diffuser discharge velocity of 1.5m/s. Air velocity was found to be varying between 0.1m/s to 0.5m/s in both sections with velocity above 0.2m/s usually occurring exactly below the diffuser location which similar to base case results which was again due to the buoyancy effect. The temperature offset occurring in velocity draft locations were upto a value of 2°C. Operative Temperature varied between 27°C to 31°C which displays the combine effect of air temperature and MRT thus moving these values beyond allowed limit. PMV was found to vary from -0.5 in the locations of velocity draft to 1.25 in other locations. Overall PMV was towards the slightly warmer side but well below the EnergyPlus predicted value. In such situation people will feel uncomfortable in the regions of higher PMV value. Thus the resultant PPD for both sections was found to vary from 13.5% to 40% with lower value was observed in the region of velocity drafts. For this diffuser discharge velocity the space can be termed as thermally not comfortable as there are warm spots which are uncomfortable areas for most occupants.

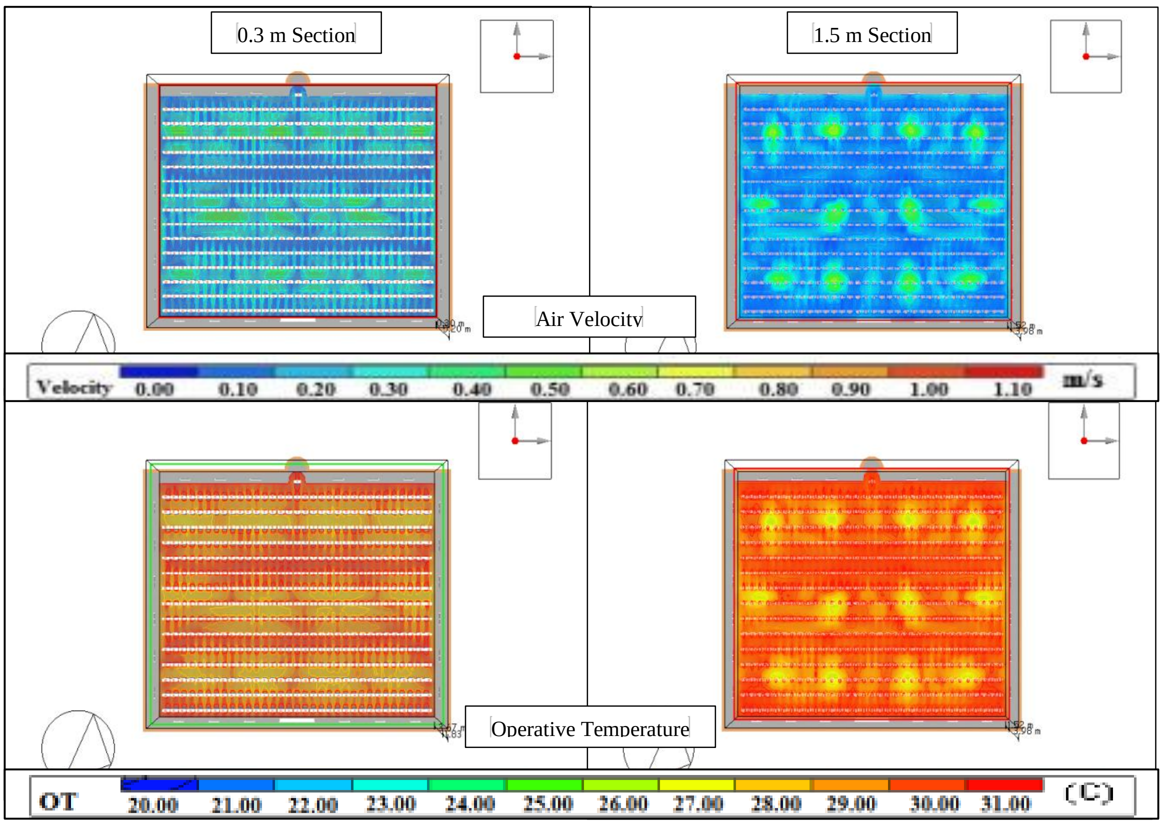


Figure 5.38: M1- Air velocity and Operative Temperature contours at sections 0.3m and 1.5 for 1.5m/s velocity

The Diffuser discharge velocity was increased and results were analyzed and it was observed that increase in diffuser discharge velocity worsens the thermal comfort situation contrary to what observed in the base case. Results of 2 m/s and 3 m/s discharge velocity showed that the air temperature value increased to the higher side and found to move towards EnergyPlus predicated value.

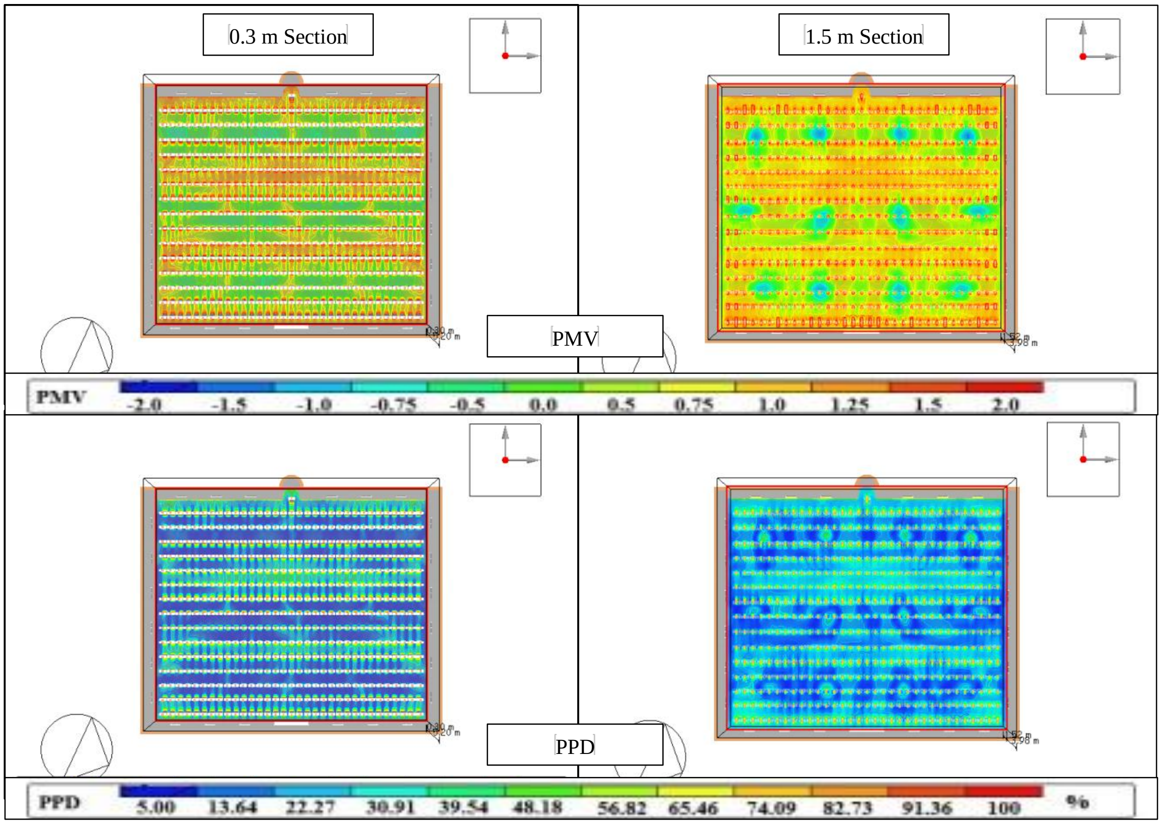


Figure 5.39: M1- PMV and PPD contours at sections 0.3m and 1.5m for 1.5m/s velocity

Lastly a diffuser discharge velocity of 3.5 m/s was used. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.40 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.41 for a diffuser discharge velocity of 3.5 m/s. Air temperature for this velocity case was more uniform and found to be varying between 26^{o}C to 29^{o}C which is very near to EnergyPlus predictions. Air velocity however was similar to base case and found to be varying between 0.1m/s to 0.4 m/s with velocity above 0.2m/s occurring exactly below the diffuser location which allowed an air temperature offset for is less than 2^{o}C as the air temperature in these areas is around 26°C and MRT 33°C, the velocity draft effect decreased significantly. Operative Temperature varied between 29°C to 31°C with 31°C being predominant which again worsens the situation. As a result PMV was in the range of 0.5 which mostly occurred in the regions of velocity drafts to 1.5 in the occupied zone. The resultant PPD was found to vary from 13.5% to around 65% with 13.5% occurring in the regions of draft.

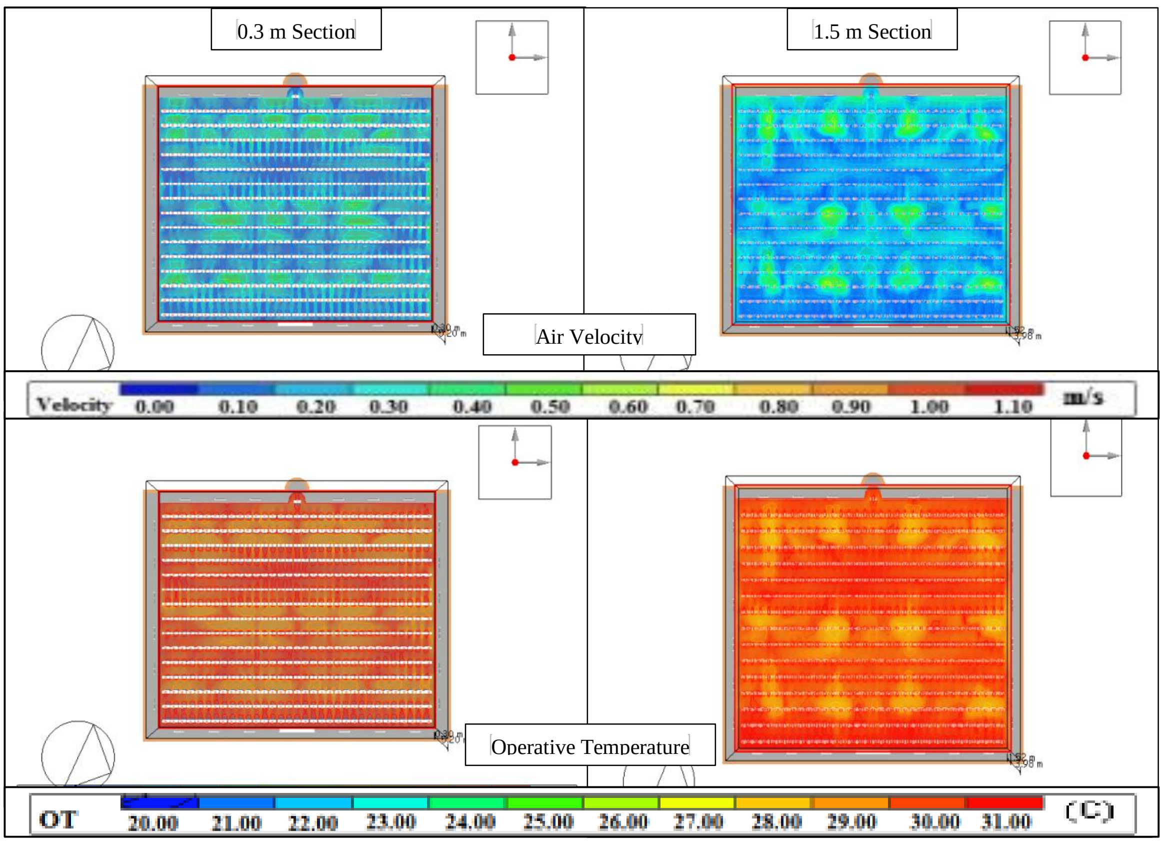


Figure 5.40: M1- Air velocity and Operative Temperature contours at sections 0.3m and 1.5m for 3.5m/s velocity

For this diffuser discharge velocity the space can be termed as thermally not comfortable and overall situation deteriorated compared to the previous velocity cases. Four-way diffusers which are known for uniform mixing of air, performed almost similar to

EnergyPlus predictions except for velocity variation. Although for the base case model higher diffuser discharge velocities yielded better results but it was contrary at the new operation strategy. This diffuser configuration works towards achieving uniform comfort values by utilizing forced convection method and does not utilize the buoyancy effect and negative pressure created towards return air diffusers. It should be remembered that only the occupied zone need to be conditioned to achieve thermal comfort for the occupants but not for whole space.

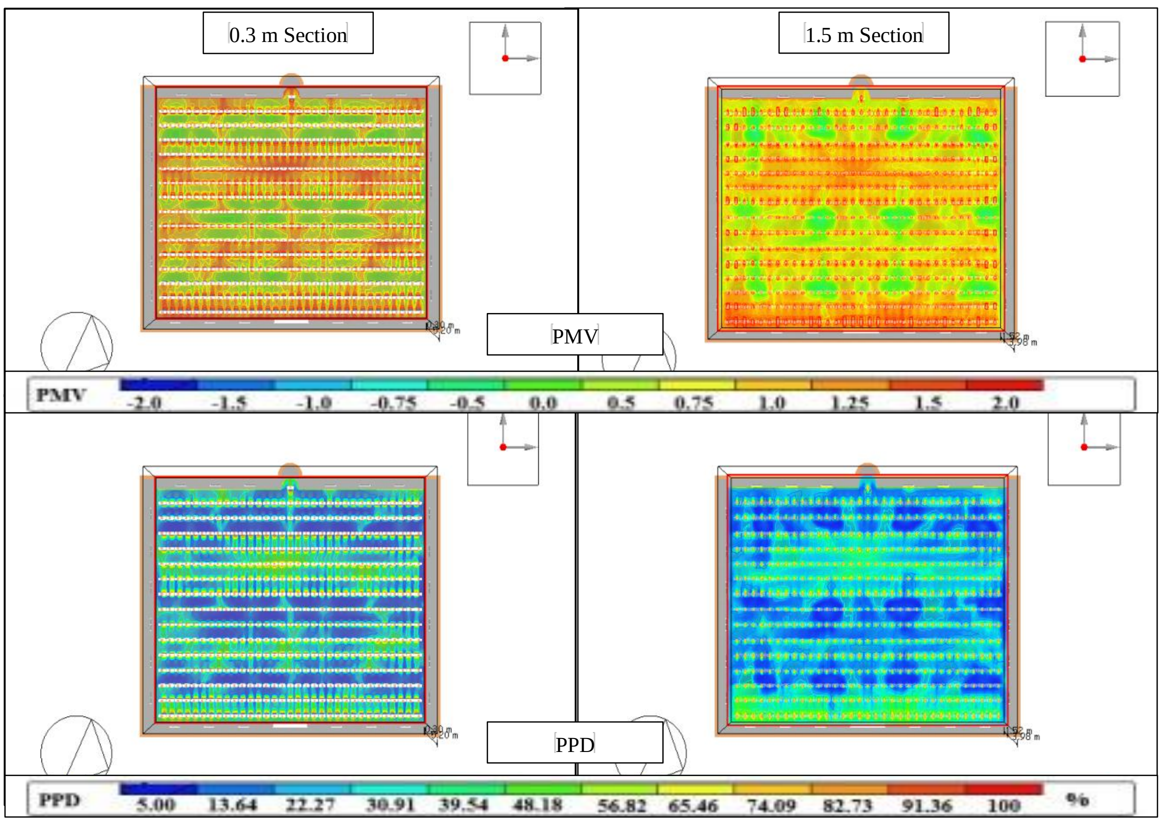


Figure 5.41: M1- PMV and PPD contours at sections 0.3m and 1.5m for 3.5m/s velocity

5.5.1.2 CBAD with linear/slot supply diffusers and Ceiling Return (M2)

M2 air distribution scheme was simulated for thermal comfort performance. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.42 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.43 for a diffuser discharge velocity of 1.5m/s. Air velocity was found to be varying between 0.1m/s to 0.5m/s in the occupied zone with velocity above 0.2m/s mostly occurring exactly below the diffuser locations which resulted in temperature offset occurring in these locations upto a value of 2^{o}C. Operative temperature for this velocity was found to be varying between 26^{o}C to 29^{o}C in the occupied zone with 24^{o}C being predominant. Operative temperature variation of 28- 29^{o}C was mostly observed to occur in the middle of the occupied zone, below the return diffuser area. These values of air temperature are acceptable but are on the warm side.

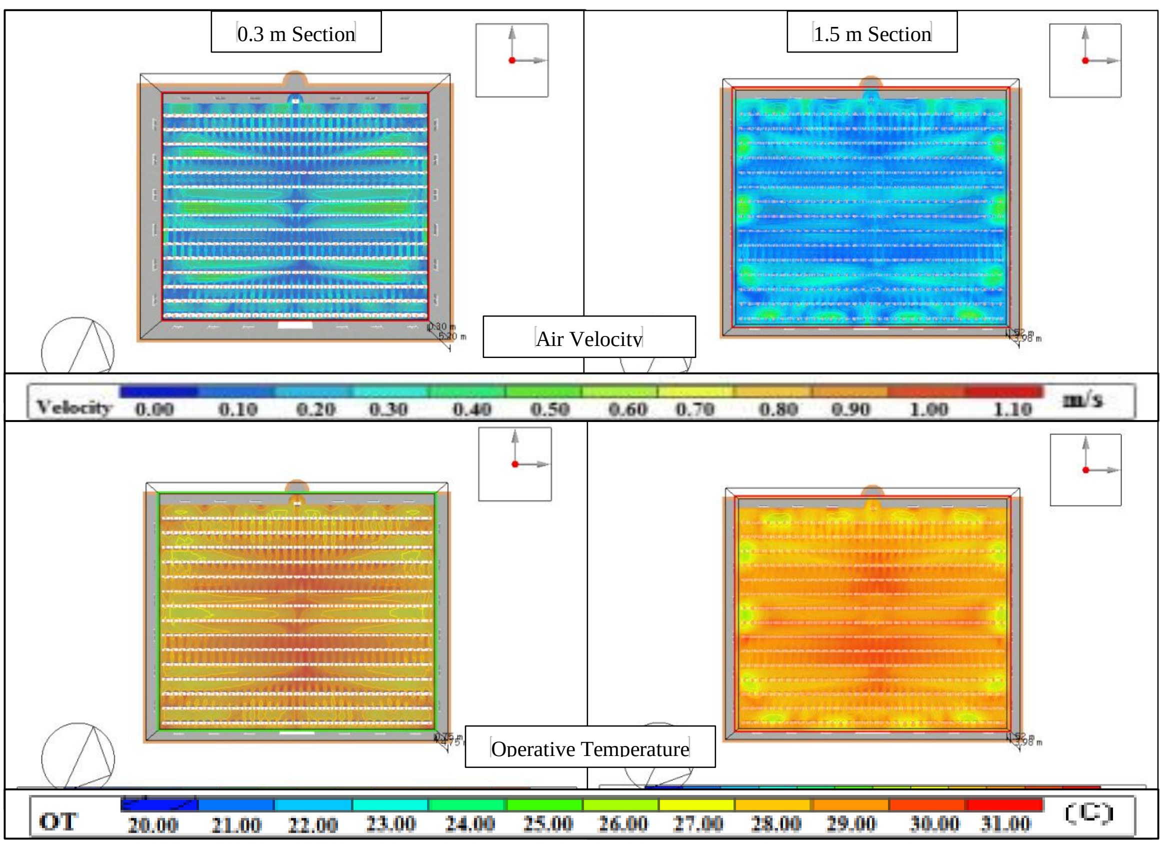


Figure 5.42: M2- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

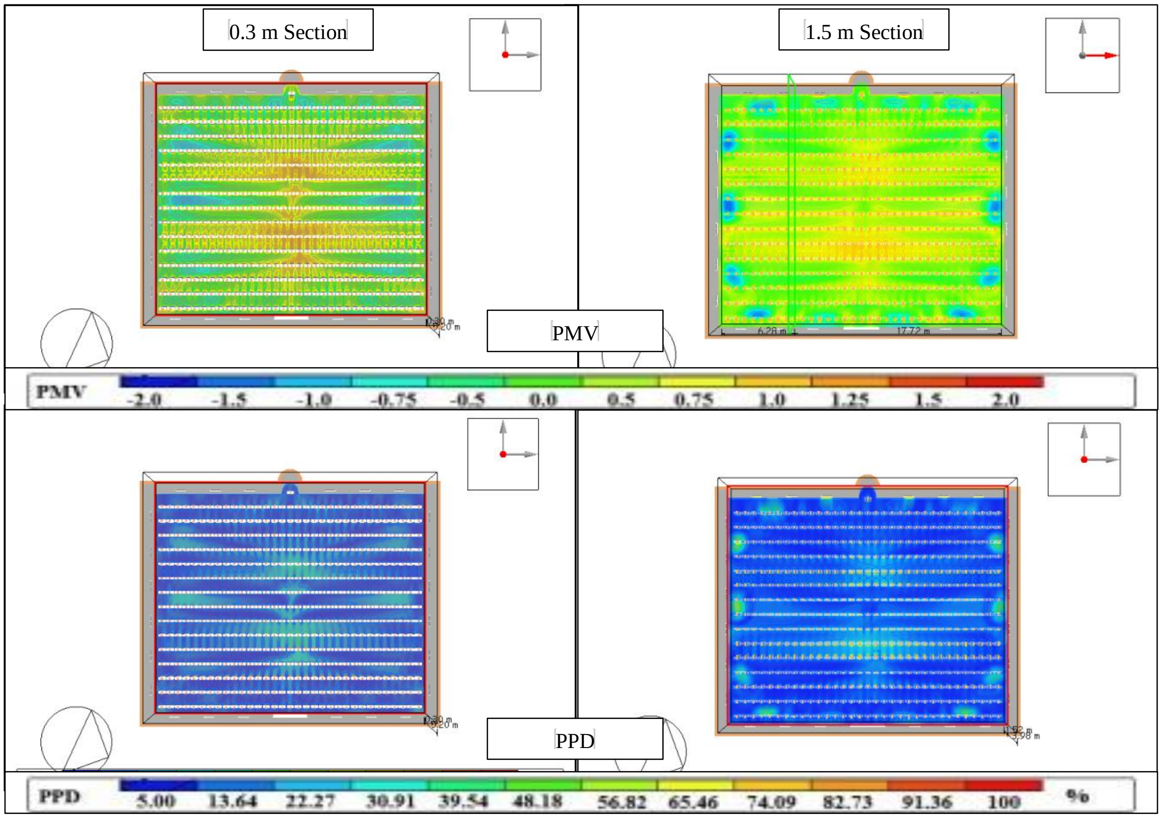


Figure 5.43: M2- PMV and PPD contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

PMV was found to vary from -0.7 at the perimeter region below the supply diffuser locations to 1.0 in the middle region. Even though the temperatures were toward slightly warmer side but the overall PMV value was around 0.5 which is acceptable. Thus PPD in the occupied zone was found to vary from 5% at the perimeter regions to 30% in the middle of the occupied zone where higher PMV values were observed. For this diffuser discharge velocity, the space can be termed as thermally comfortable, and the overall situation is much improved compared to M1 model with same velocity. One important thing noticed here was that in base case results for this velocity was on cool side and here it is performing much better mostly due to buoyancy effect. However, slightly warm spots still exist and need to be take care of, so discharge velocity was increase to enhance the convection effect.

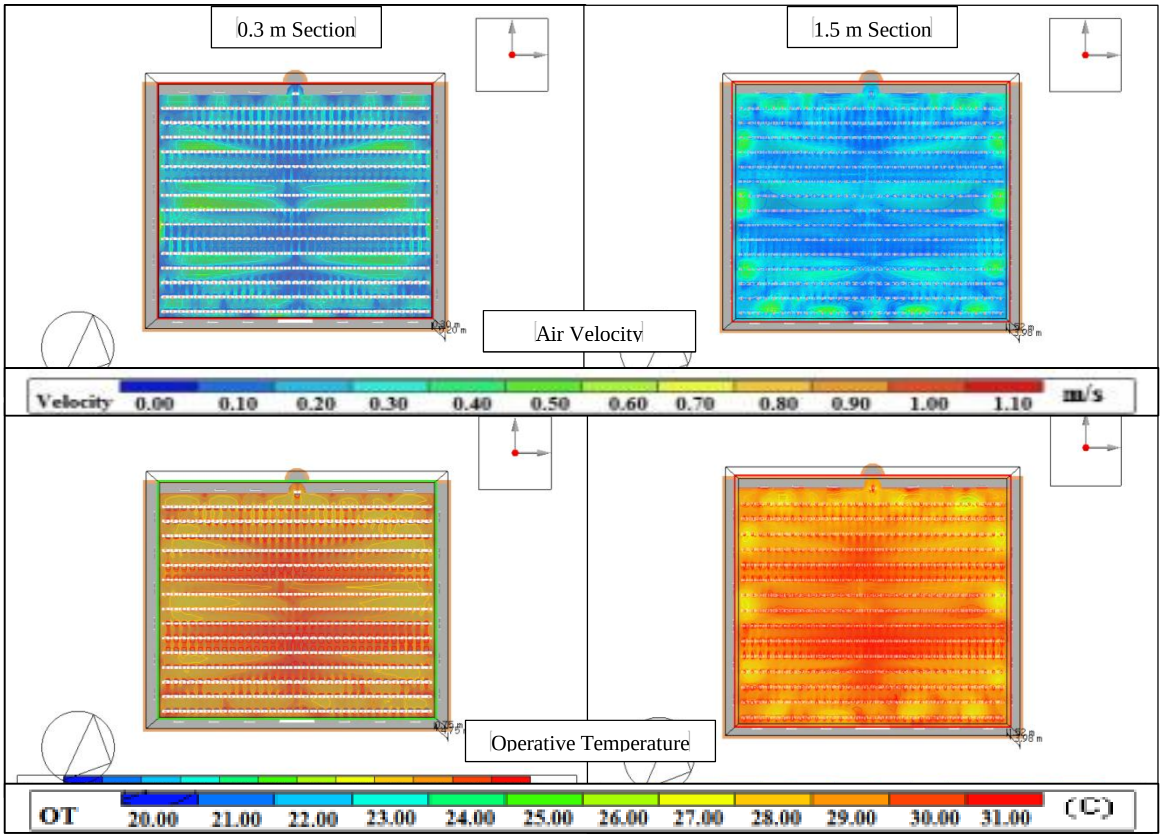


Figure 5.44: M2- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

A velocity of 2 m/s was tested in order to assess the improvement in the situation and results showed similar values for air temperature, air velocity, and operative temperature but the area of the slightly warm spot increased which resulted in consequent increase in PPD in the occupied zone. And the same phenomenon was observed with further increase of diffuser discharge velocity to 3 m/s. The supply diffuser discharge velocity was increased to 3 m/s and results show that the problem of slightly warm spots amplified. A further increase in supply diffuser discharge velocity to 3.5m/s was tested.

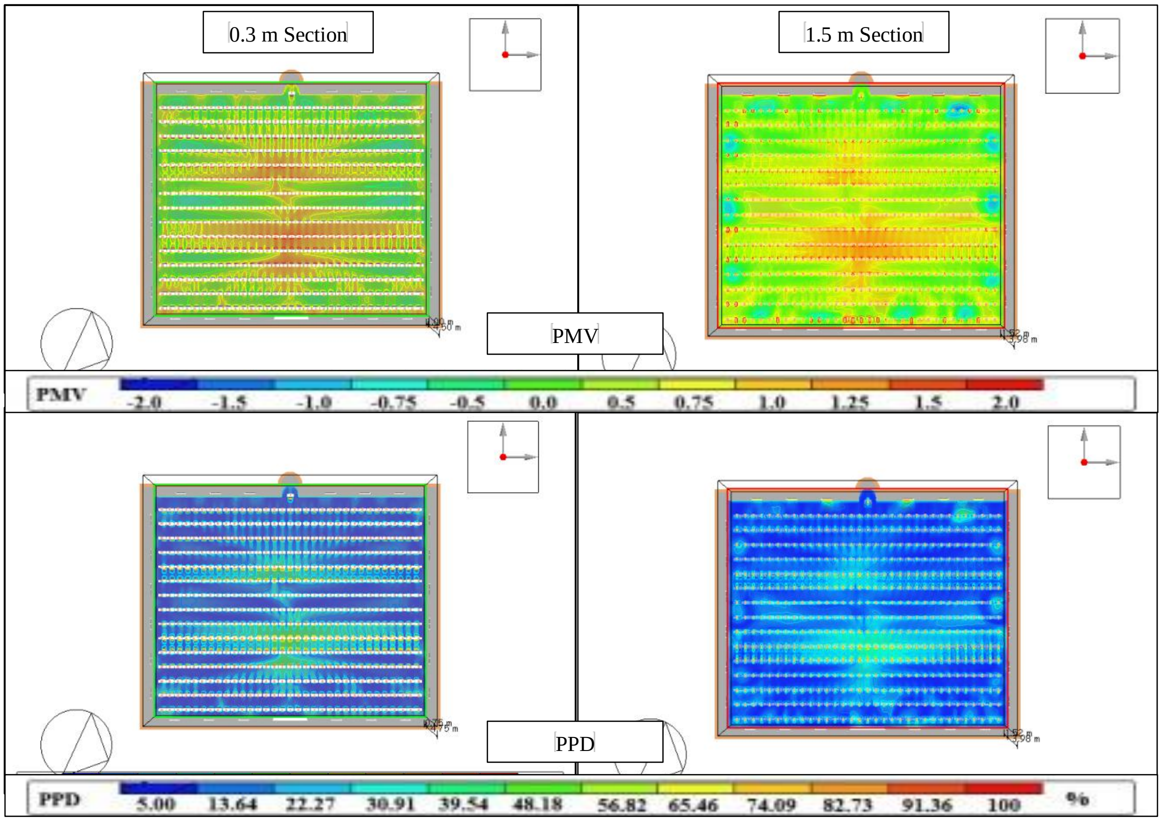


Figure 5.45: M2- PMV and PPD contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.44 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.45 for a diffuser discharge velocity of 3.5 m/s. Air velocity was similar to base case and found to be varying between 0.1 m/s to 0.4 m/s in both sections with 0.25 m/s as average velocity value with very minimal temperature offset occurring in the elevated velocity regions. Operative Temperature was found varying between 27°C to 30°C which was worst in all velocity cases of this model. Consequently PMV was found to vary from 0.0 at the perimeter region to 1.25 in the middle region with 0.5 being the average value. It was observed that lower PMV values were due the low temperature value along with velocity drafts. The resultant PPD for both sections was found to vary from 5% to 40% with maximum areas of 13.5%. Although this value is very near to the required 20% or less PPD but for this diffuser discharge velocity the space can be termed as thermally not comfortable. M2 performed better than M1 but still it has slightly warm spots and these spots were less in area at low discharge velocity thus showing dominant buoyancy effect. However the slightly warm areas needed to be reduced, So model M3 with return on wall near to the occupied zone was test to enhance the situation.

5.5.1.3 CBAD with linear/slot supply diffusers and Wall Return (M3)

This model performed well in achieving thermal comfort in the occupied zone when simulating for continuous case of HVAC operation. Again the same setup is used with changes in temperature boundary condition that were obtained for intermittent operation from EnergyPlus simulations. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.42 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.47 for a diffuser discharge velocity of 1.5m/s. Air velocity was found to be varying between 0.1m/s to 0.5m/s in both sections with 0.3 m/s average velocity in the entire occupied zone. Thus the temperature offset occurring in locations with velocity higher than 0.20 m/s was observed to be less than 2^{o}C. Operative Temperature was varying between 25^{o}C to 30^{o}C with very few spots of 25^{o}C below the supply diffuser location. This increase in operative temperature was gradual towards the return. As a result PMV was found to vary from -1.0 at the perimeter region where there was velocity draft to 1.0 towards the return diffuser. In the entire occupied zone PMV was 0.5 - 0.7 which is encouraging. The resultant PPD in the occupied zone was found to vary from 13.5% in the middle of the occupied zone to 40% in the regions below diffusers and near return diffuser.

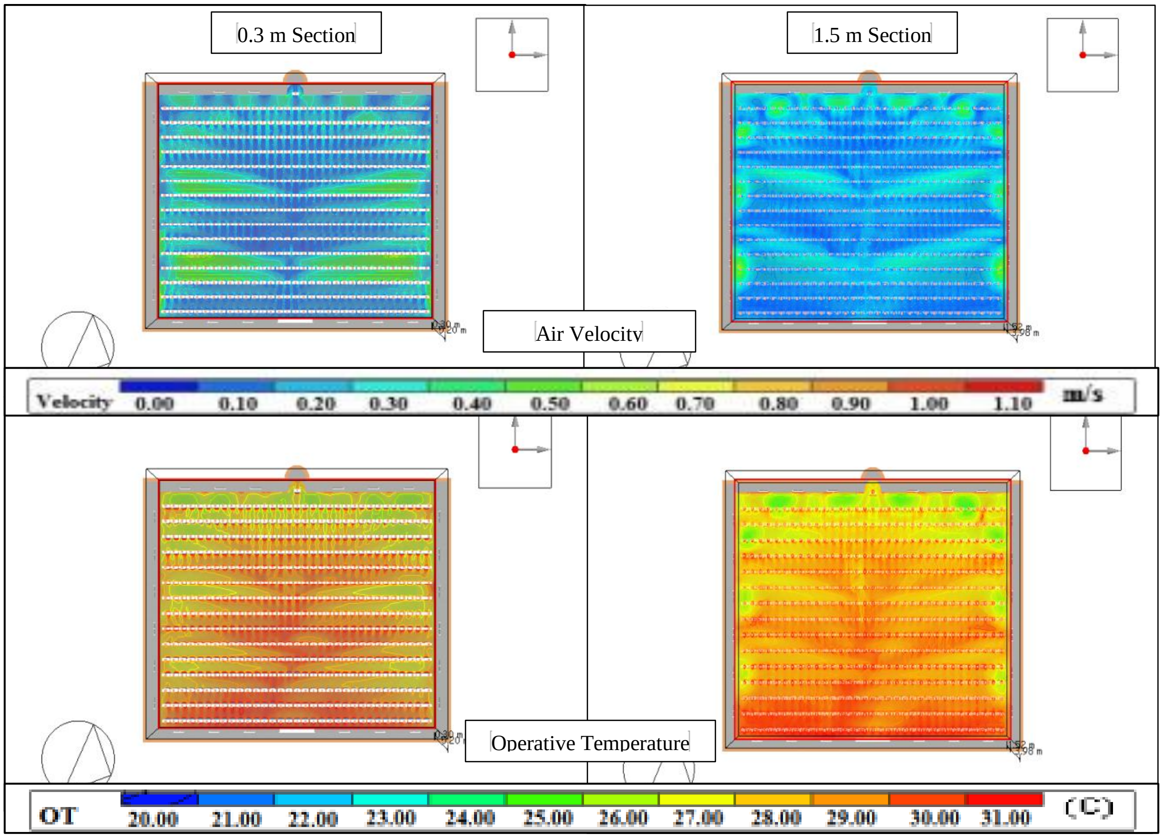


Figure 5.46: M3- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

It goes on to show that the space is mostly comfortable due to a balance between buoyancy and convection effect, but with temperature and velocity drafts occurring below diffusers. Here was a new problem of slightly cool and slightly warm spots appearing in the occupied zone. From the base case simulation it was learnt that the slightly cool spots can be avoided by increasing the velocity of discharge of the diffuser. The draft regions need to be reduced, which was done by increasing the supply diffuser discharge velocity. For a diffuser discharge velocity 2 m/s the situation was more or less the same. So supply diffuser discharge velocity was increased to 3 m/s. with this increase the comfort regions were more uniform with PMV improving to -0.7 at the perimeter region to 0.5 in the middle region but the slightly warm spots were still available near the return diffuser. The resultant PPD was found to vary from 13.5% in most regions to 30% in the region below diffusers and near return diffuser.

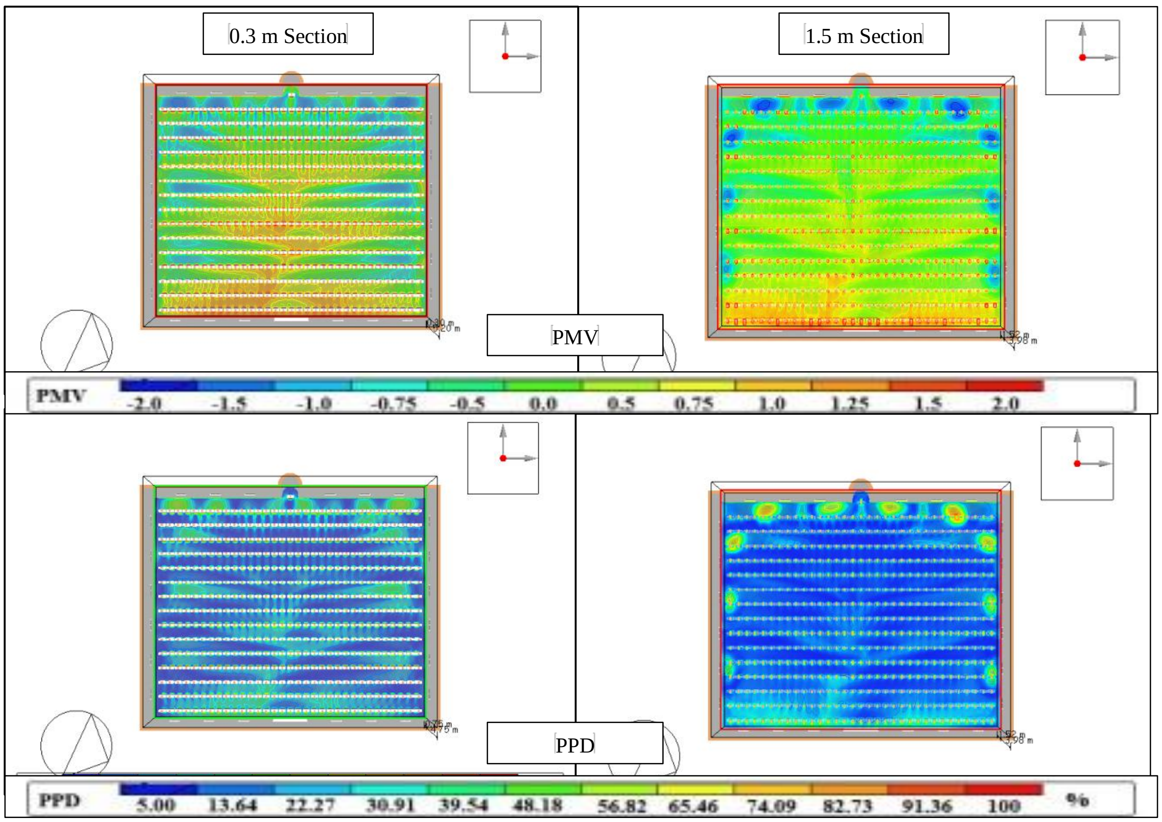


Figure 5.47: M3- PMV and PPD contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

Lastly a diffuser discharge velocity of 3.5 m/s was used to ensure more reduction of slightly cool and slightly warm spots. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.48 and resulting PMV and

PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.49 for a diffuser discharge velocity of 3.5m/s.

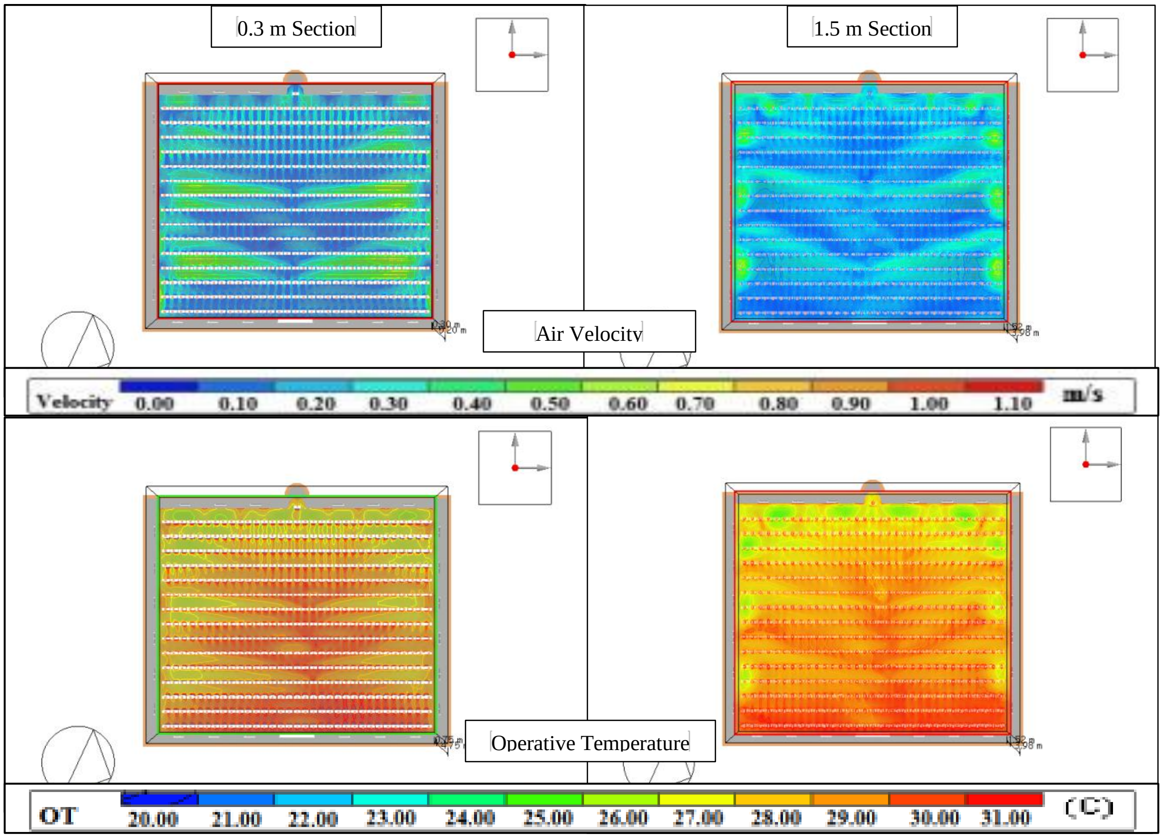


Figure 5.48: M3- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

Air velocity contours did improve a little, varying between 0.1m/s to 0.4m/s in both sections with velocity above 0.3m/s usually occurring exactly below the diffuser location but in much reduced area, indicating that there would be temperature offset 1ºC or less. Likewise, Operative Temperature was varying between 25ºC to 30ºC with 28ºC prevalent in the whole space. Thus PMV was seen falling between -0.7 at the perimeter region where velocity draft was observed to 0.5 in the middle region 0.7 in the region near return. The resultant PPD in the occupied zone was found to vary from 13.5% to 30% in the region below diffusers and near return demonstrating that space is almost thermally comfortable for 80% or higher occupants but with slightly warm spots still appearing. It was observed that higher diffuser discharge velocity improved thermal comfort, by increasing the balance between buoyancy and convection effects but the slightly warm spot problem still existed which prompted the next model.

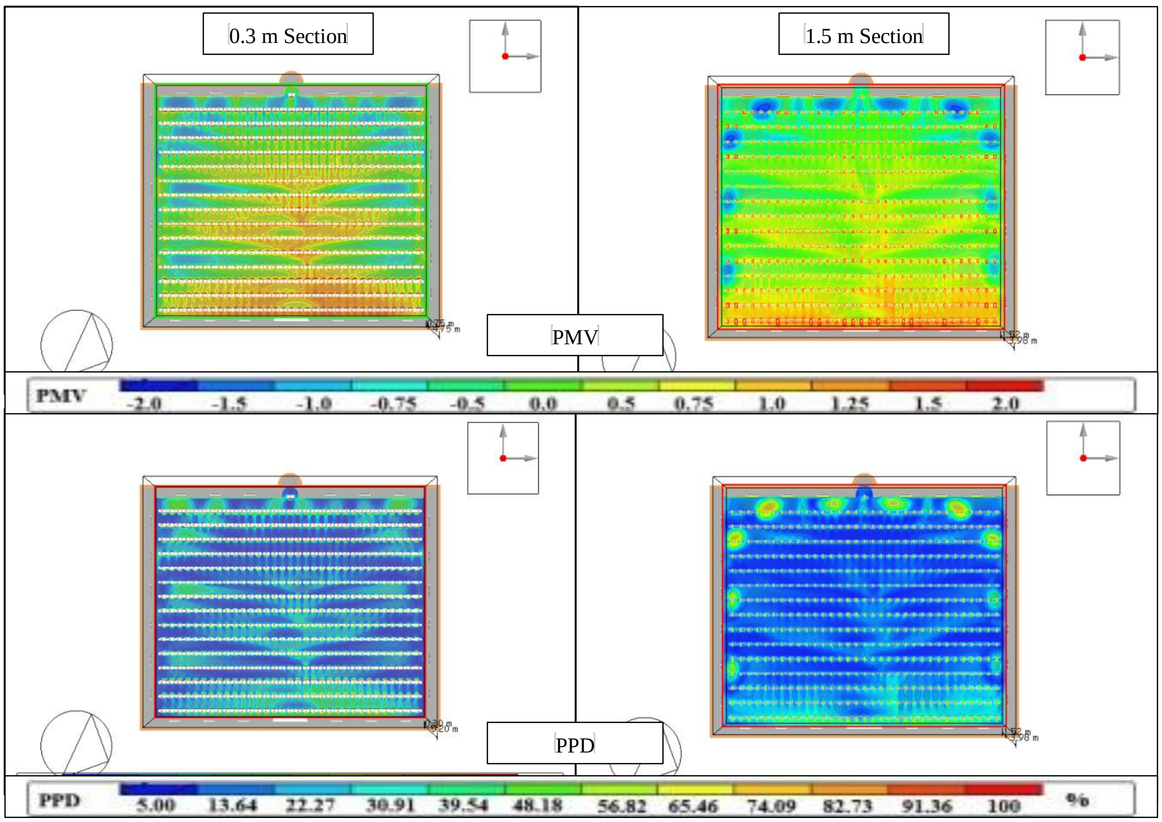


Figure 5.49: M3- PMV and PPD contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

5.5.1.4 CBAD with linear/slot supply diffusers and Wall Return (M3-1)

As a part of sensitivity analysis of M3, M3-1 was created to reduce the slightly warm spots near the door. This model was simulated for only one diffuser discharge velocity which is 3.5 m/s. Resulting Air velocity and Operative Temperature contours at sections

0.3 m and 1.5 m are shown in Figure 5.50 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.51 for a diffuser discharge velocity of 3.5m/s.

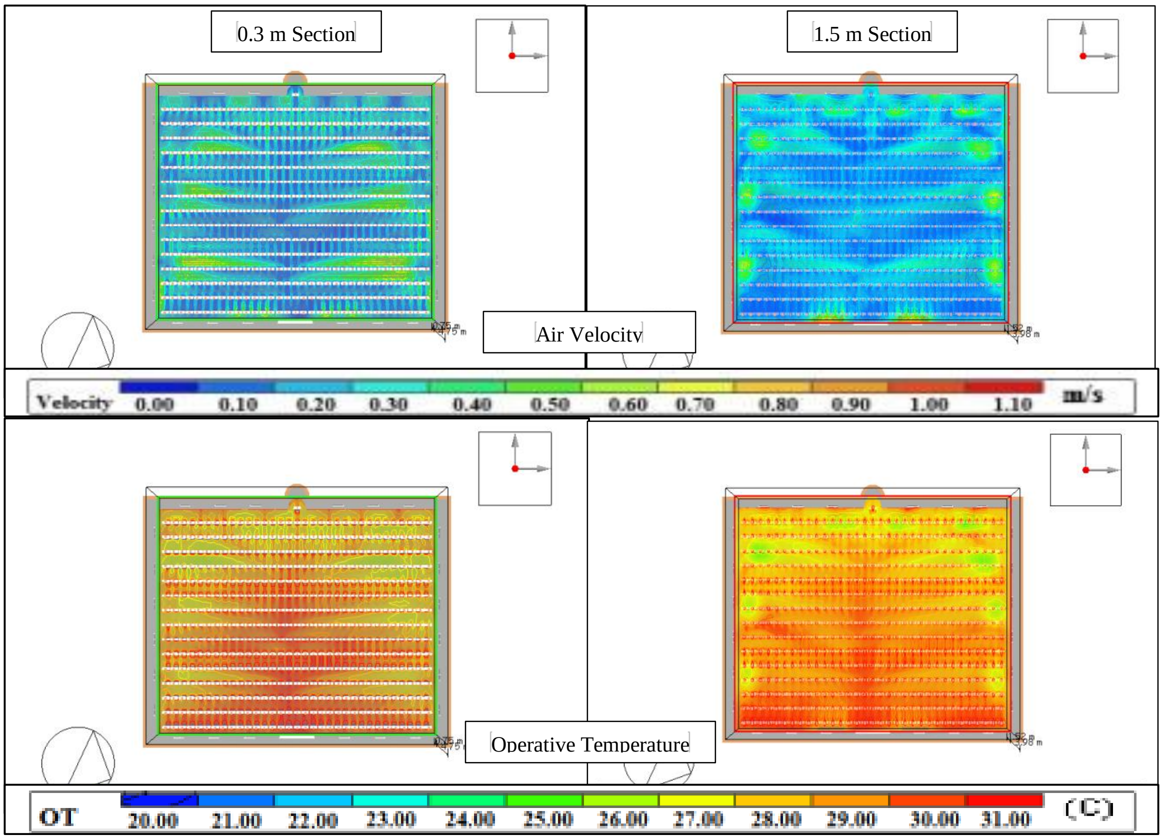


Figure 5.50: M3-1- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

Air velocity was more uniform, varying between 0.1m/s to 0.4m/s in occupied zone with velocity above 0.3m/s usually occurring exactly below the diffuser location but in much reduced area, indicating that there would be temperature offset 1^{o}C or less. Likewise, there was elevated velocity observed near the return diffuser which is now in the occupied zone at a value of 0.5 m/s. Operative Temperature was varying between 25^{o}C to 30^{o}C with increased prevalence of 28^{o}C in the whole space. Thus PMV was observed to be between 0.0 at the perimeter region where velocity draft was observed to 0.5 in the middle region 0.7 in the region near return. Areas with PMV 1 were not observed to occur in the occupied zone. The resultant PPD in the occupied zone was found to vary from 13.5% to 30% in the region below diffusers and near return demonstrating that space is almost thermally comfortable for 80% or higher occupants. It was observed that change in return diffuser location did affect the thermal comfort status at high diffuser discharge velocity and removed the slightly warm spot problem that existed in previous models. However, there is a concern over its practicality. Thus, further air distribution strategies were analyzed before deciding upon any conclusion.

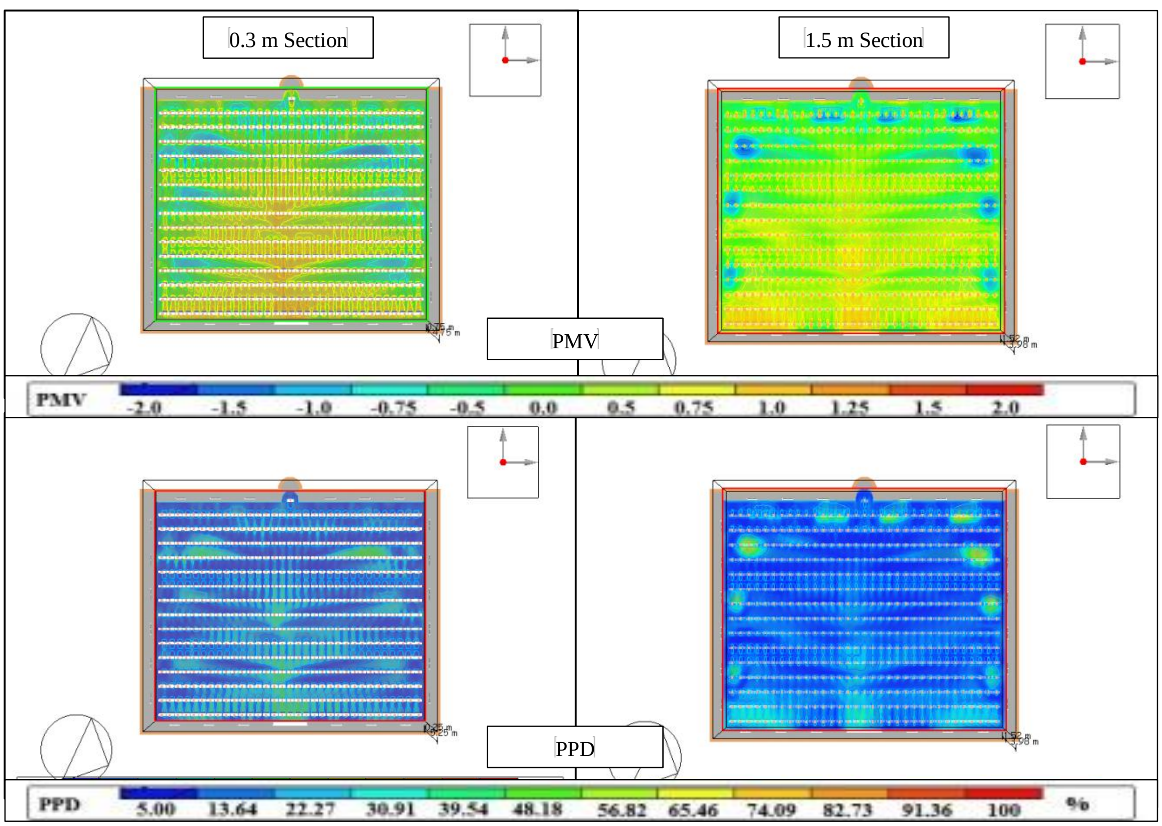


Figure 5.51: M3-1- PMV and PPD contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

5.5.2 Through-Wall Air Distribution (TWAD) (M4 and M5)

This air distribution strategy had worst performance as far as thermal comfort goes in the base case simulations with most velocities shows sever cold spots in the occupied zone. This strategy has the potential to solve the slightly warm spots problem.

5.5.2.1 TWAD with linear/slot supply diffusers and return on Ceiling (M4)

The setup of this model is same as in base case simulations. The velocities at the diffuser discharge considered were 1.5 m/s, 2 m/s, 3 m/s and 3.5m/s. As observed in base case simulations, there were cool spots at the throw area of the diffusers. For the diffuser discharge velocities 1.5 m/s and 2 m/s had similar observations. PMV in the case of 1.5 m/s velocity varied between -1.5 at the throw area to 0.5 in the middle of the occupied zone. The corresponding PPD in the throw areas was at 100% and in middle of the zone at 5% which is improved situation in the middle of the occupied zone compared to base case results. In case of 2 m/s diffuser discharge velocity similar observations were made with PMV in the throw areas increased to -1.25 with consequent decrease in the PPD value to 80%. This goes to show that PMV and PPD are improving with increase in the diffuser discharge velocity. The diffuser discharge velocity was increased to 3 m/s. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.52 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.53 for a diffuser discharge velocity of 3 m/s. Air velocity was found to be varying between 0.1m/s to 0.6m/s in occupied zone with velocity above 0.3m/s occurring at the throw area which allowed a temperature offset occurring in locations of velocity draft upto a value of 3^{o}C as the difference between air temperature and MRT in most locations is around 9^{o}C resulting in larger temperature offset value the occupied zone. Operative Temperature also resulted towards comfort region, varying between 24ºC to 29ºC in the occupied zone and to 24- 26ºC occurring in the throw areas mainly influenced by air temperature. PMV was found to vary from -1 at the throw area in the occupied zone to 0.50 in rest of the occupied zone. As a result PPD 30% in the velocity draft areas to 13.5% in most of the occupied zone. For this diffuser discharge velocity the space can be termed as thermally comfortable and it improved significantly compared to situation in previous models. Thus the diffuser discharge velocity was further increased to 3.5 m/s.

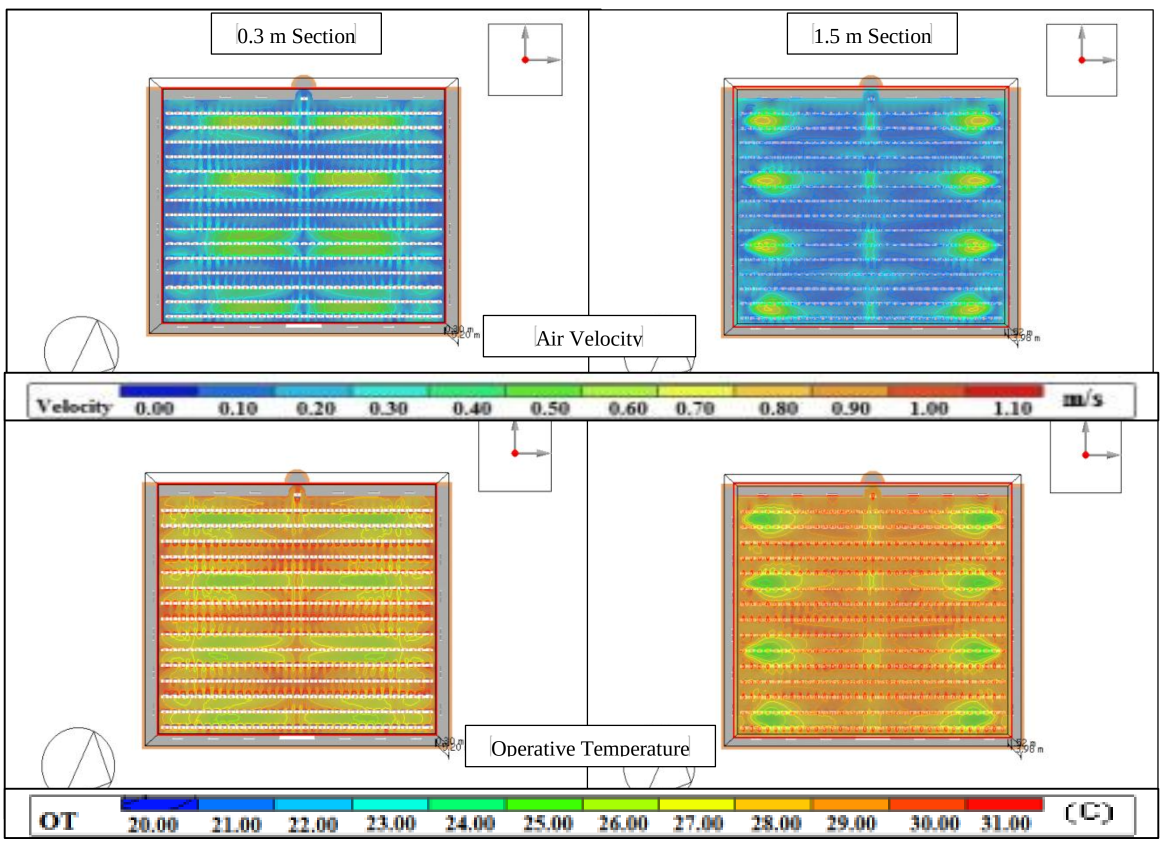


Figure 5.52: M4- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 3 m/s velocity

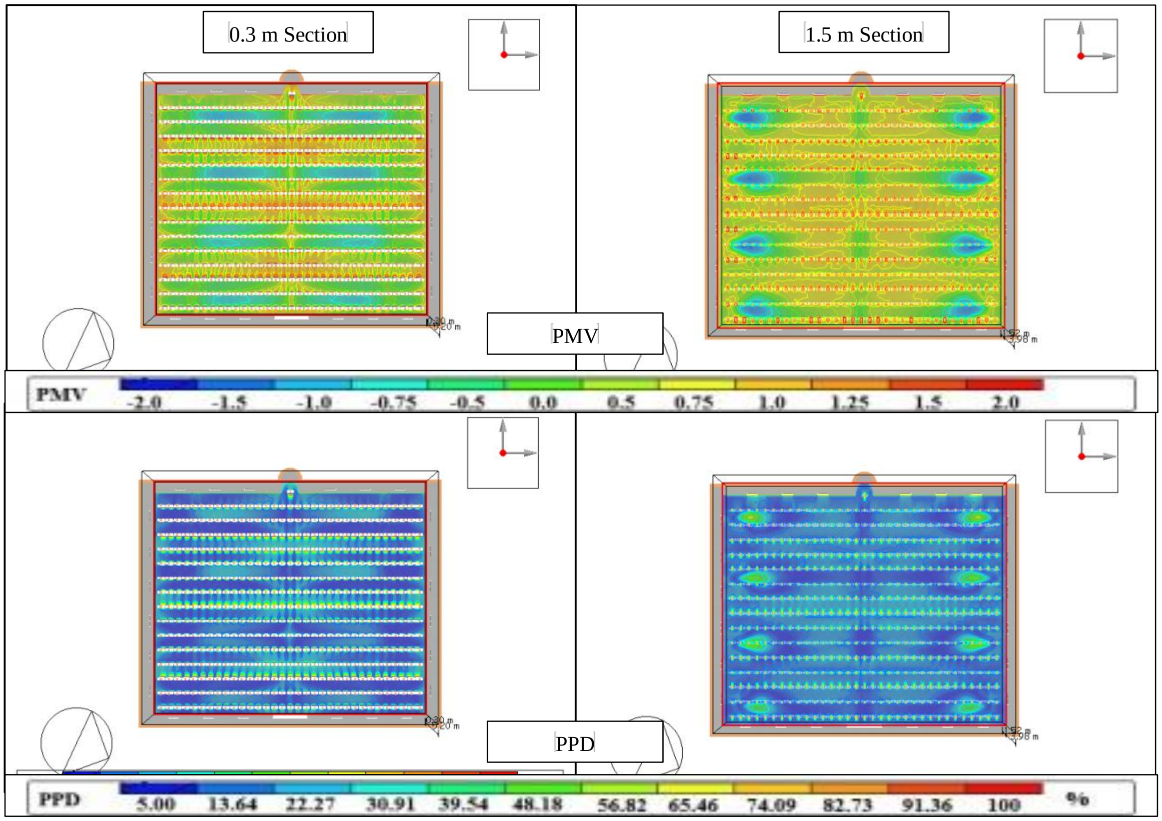


Figure 5.53: M4- PMV and PPD contours at sections 0.3 m and 1.5 m for 3.0 m/s velocity

Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.54 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.55 for a diffuser discharge velocity of 3.5 m/s. Observations made here were very intriguing. Air velocity was again observed to have changed drastically and was found to be varying between 0.2 m/s to 0.6 m/s with 0.4 m/s occurring predominantly in the occupied zone causing temperature offset in almost all locations upto at least a value of 3°C. Operative Temperature also resulted towards slightly warm region, varying between 28°C to 30°C in the occupied zone. There was temperature stratification observed in the occupied zone but within the allowable limits of

3^{o}C. Consequently PMV was found to vary from 0.0 to 1.25 showing the effect of velocity drafts improving the thermal comfort scenario. PPD resulted to vary around 5% to 40% in slightly warm areas which occurred at the perimeter zones. Now, these results were further analyzed to understand the reason behind these significant variations in the occupied zone at this diffuser discharge velocity. It turns out that there was a short circuit happening towards the return diffuser due to the elevated loads in the complete space. Never the less this model performed exceptionally well compared to previous models to change in loads.

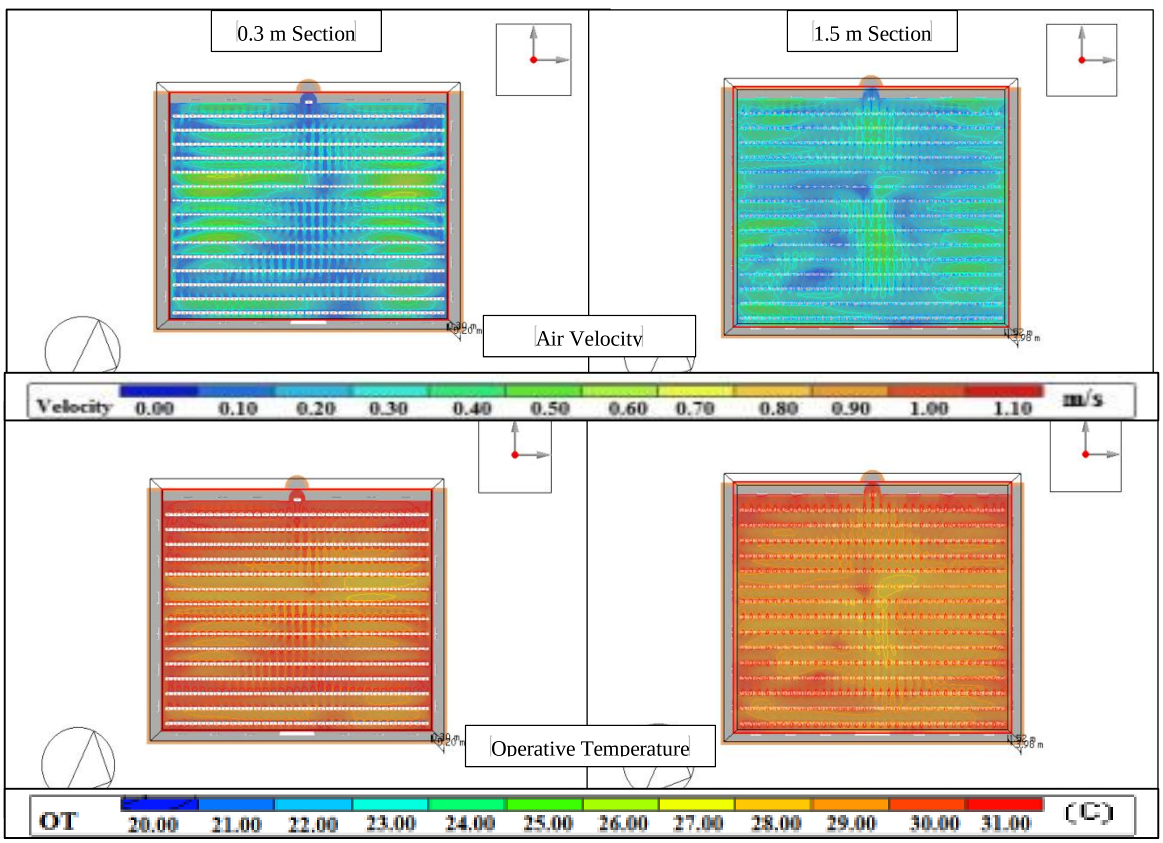


Figure 5.54: M4- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

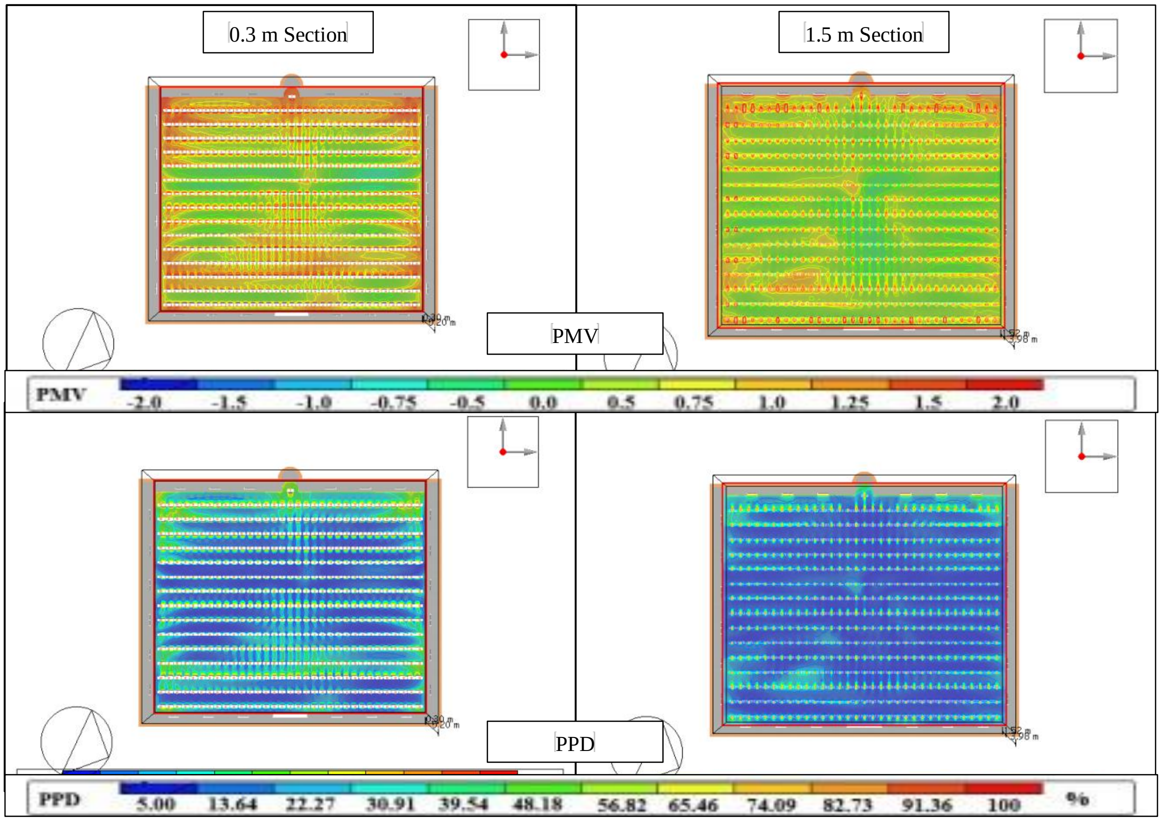


Figure 5.55: M4- PMV and PPD contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

5.5.2.2 TWAD with linear/slot supply diffusers and return on wall (M5)

Again the setup of this model remains the same as in base case simulations. The good thing about M5 compared to M4 is the more uniformity in conditioned air supply with increased number of supply diffusers and return at the wall. Similar to previous model, this model was also analyzed for 1.5 m/s, 2 m/s, 3 m/s and 3.5 m/s as the diffuser discharge velocities. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.56 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.57 for a diffuser discharge velocity of 1.5 m/s. The problem with low diffuser discharge velocities in TWAD system was there in ability to maximize the throw distance there by crating cool spots below diffuser locations by not allowing convective effect to increase to its maximum value. The same problem was observed with 1.5 m/s and 2 m/s velocity in this operation strategy as well.

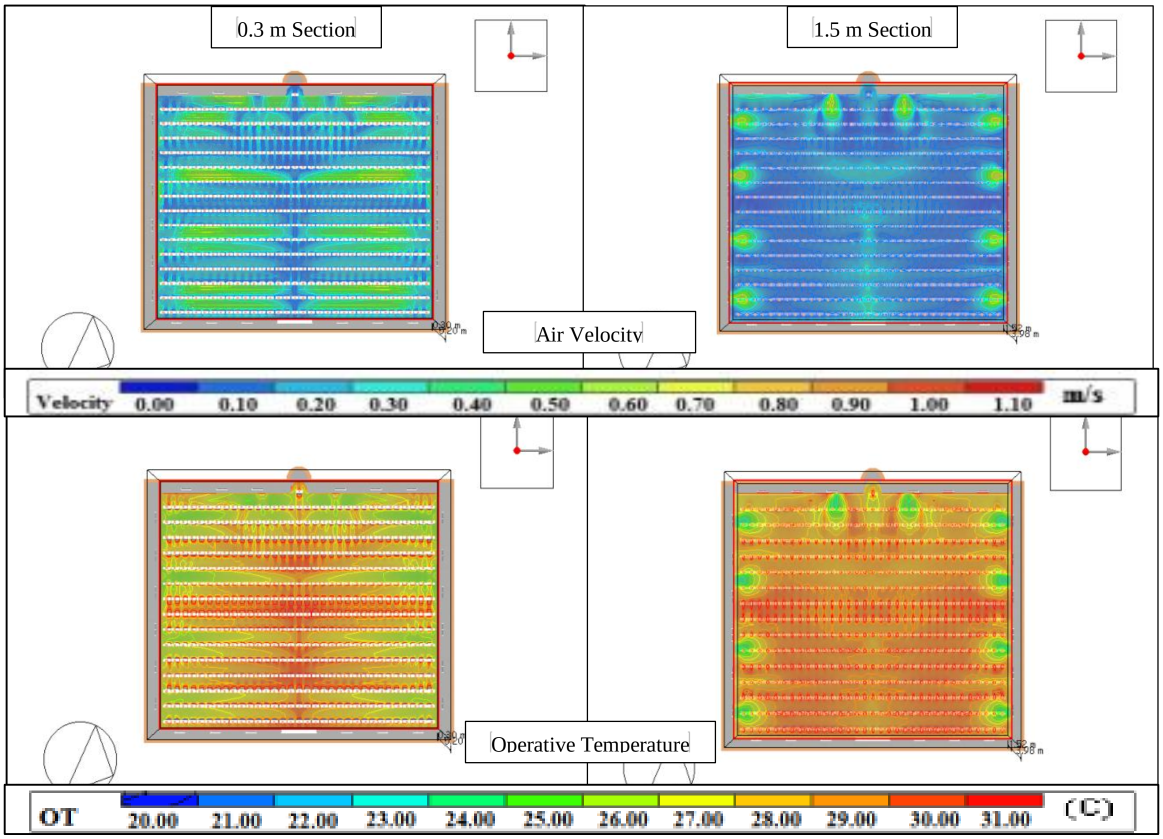


Figure 5.56: M5- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

Air velocity contours were found to be varying between 0.1m/s to 0.5m/s with 0.5 m/s velocity occurring in the region below diffusers in the occupied zone because of the buoyancy effect. There will be temperature offset occurring in regions of high velocity upto a value of 4^{o}C since MRT and air Temperature difference is around 11^{o}C. Operative Temperature also resulted towards cool value, varying between 26^{o}C to 29^{o}C in the occupied zone influenced by both air temperature and MRT. PMV was found to vary from -2, which is not desirable in any situation, predominantly occurring in the regions below the supply diffuser locations, but the overall PMV in the occupied zone was varying between 0.00 and 0.75 which is good from comfort point of view. The PPD results in these velocities and temperature drafts location showed 100% dissatisfaction areas which are not desirable in any situation but overall PPD was 13.5% excluding draft locations. Results of this model compared to M4 model for same diffuser discharge velocity are better because of the extra diffusers which reduced the volume flow rate in the diffusers.

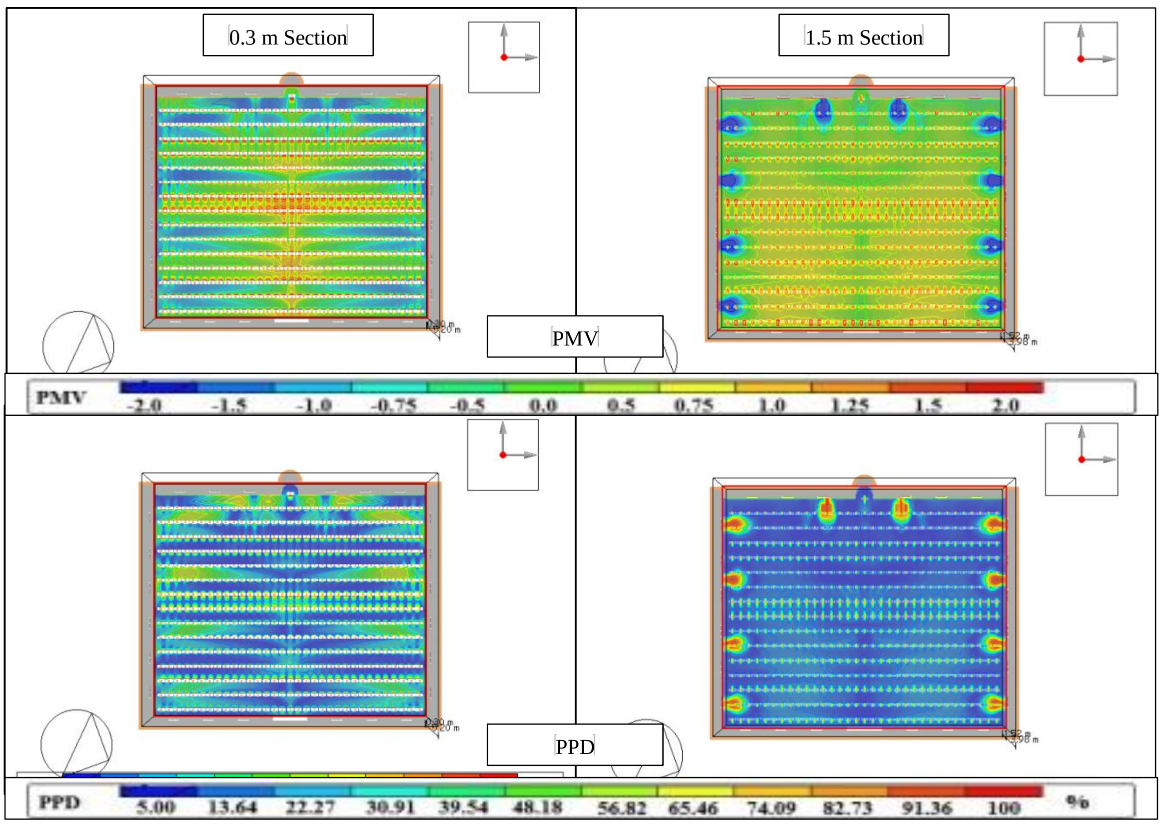


Figure 5.57: M5- PMV and PPD contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

The diffuser discharge velocity was increased to 2 m/s which gave similar results compared to 1.5 m/s velocity case and only increased the throw distance of the diffuser. Velocity 3 m/s was used as diffuser discharge velocity. The comfort status drastically changed towards uniform comfort with cool spots disappearing and giving good results for this velocity.

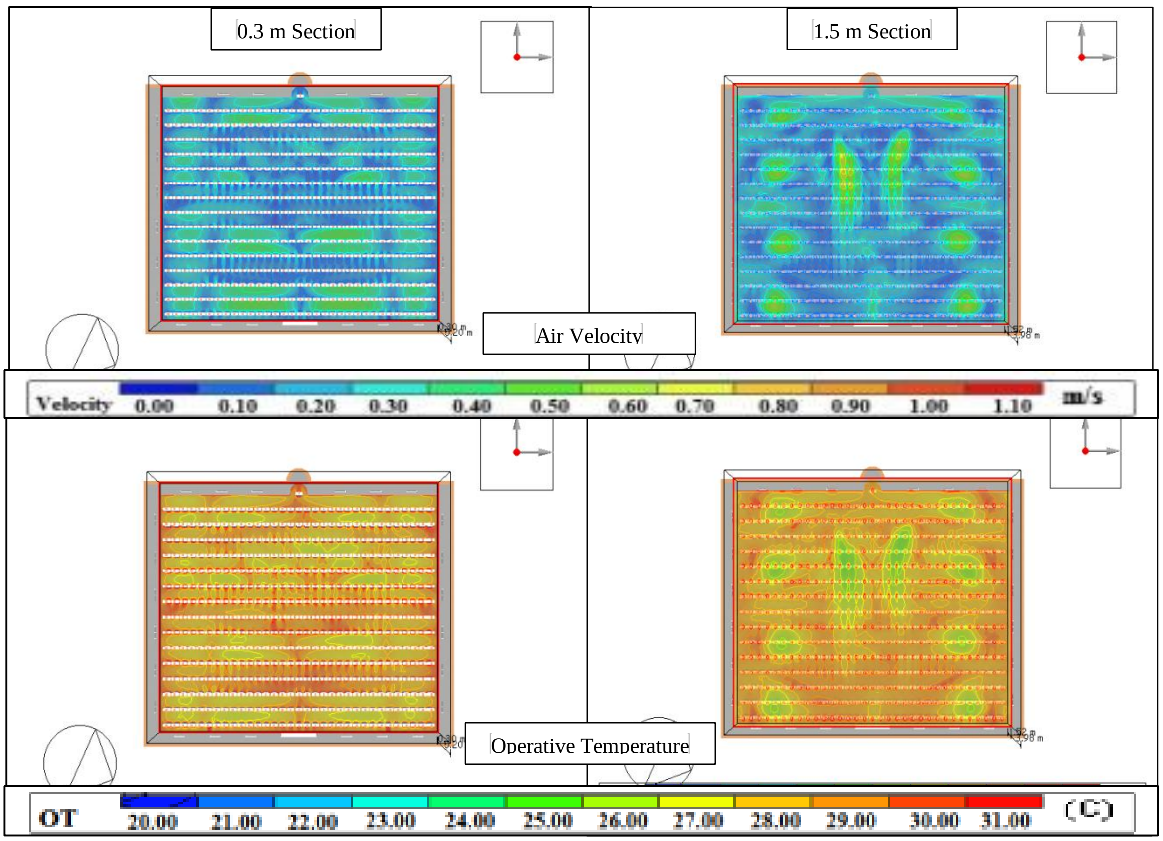


Figure 5.58: M5- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

Lastly a diffuser discharge velocity of 3.5 m/s was tested. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.58 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.59 for a diffuser discharge velocity of 3.5 m/s. Air velocity contours were found to be varying between 0.1m/s to 0.5 m/s in the occupied zone with very few spot of velocity above 0.4 m/s which show the effect of increasing the discharge velocity. Although there was occurrence of temperature offset in most locations upto a value of 2^{o}C in locations having velocity above 0.4 m/s which acceptable when the overall situation is assessed.

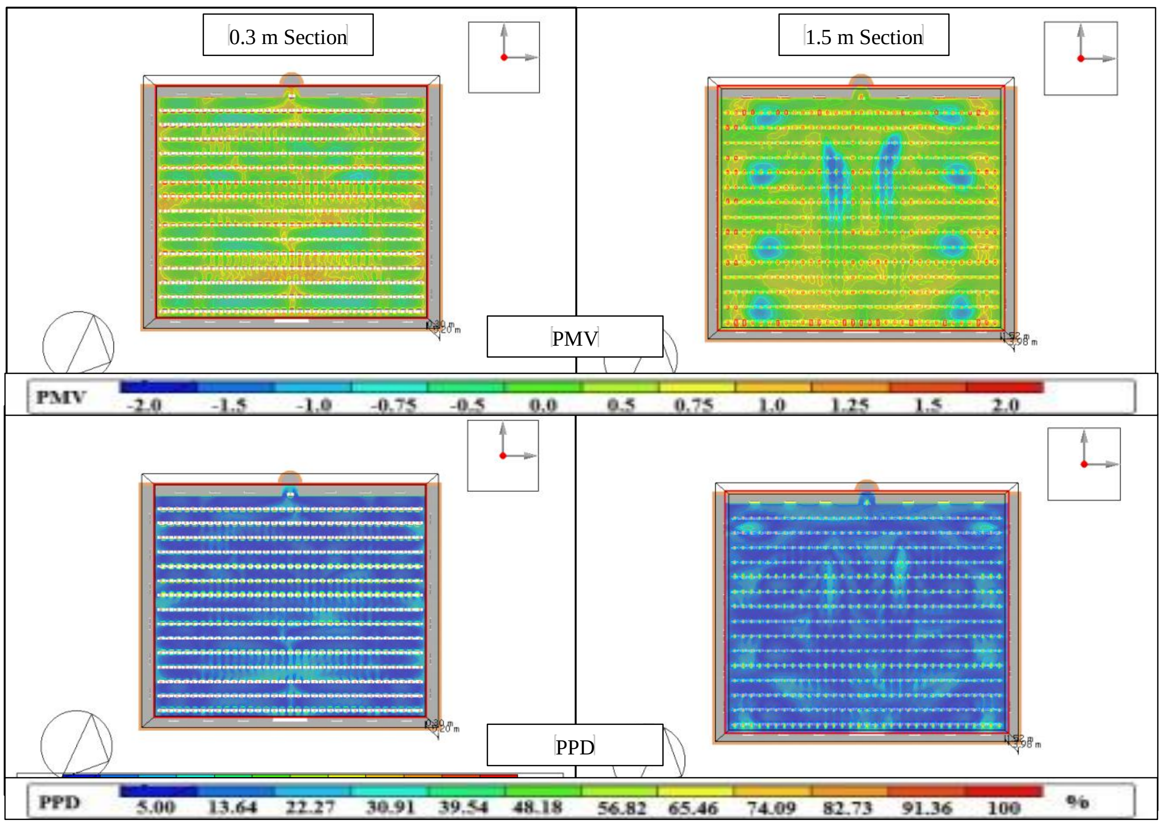


Figure 5.59: M5- PMV and PPD contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

PMV was found to vary from -0.5, which occurred mostly in the draft regions to 0.75 with 0.0 to 0.5 being predominant in the occupied zone. Operative Temperature also resulted in comfort zone with variation between 26^{o}C to 28^{o}C in the occupied zone influenced by both air temperature and MRT There was hardly any slightly warm spot observed with this diffuser discharge velocity. This resulted in PPD of 5% to 13.5 % overall with few spots of 30% in the location of the drafts, achieving the required 20% PPD criteria. One important thing noticed here is that this value of velocity caused the air to mix more uniformly. This model performed exceptionally well compared to previous models even though the MRT was very high. If compared to previous model (M4), there was no short circuiting of the flow occurred, the enhancing overall thermal comfort situation. However, UFAD needs to be analyzed as it known for temperature stratification, which may enhance the thermal comfort situation.

5.5.3: Under-Floor Air Distribution (UFAD) (M6, M7 and M7-1)

5.5.3.1 UFAD with linear/slot supply diffusers and return on Ceiling (M6)

The setup for this model remains unchanged compared to base case model. Again the same diffuser discharge velocities, 0.8 m/s, 1.0 m/s, 1.25m/s and 1.5 m/s were used. Since the variation in diffuser discharge velocity is very low; the results obtained are very similar in each of the velocity case without any visible change. So the results of highest velocity, 1.5 m/s are discussed here. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.60 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.61 for a diffuser discharge velocity of 1.5m/s. Air velocity was found to varying between 0.1m/s to 0.3m/s in both sections which conforms with the comfort limit and does not allow any temperature offset. Operative Temperature also resulted towards comfort region, varying between 25^{o}C to 29^{o}C in the occupied zone, and a value of 20^{o}C mainly occurring near diffusers. In the occupied zone, temperature stratification was within the allowable limit of 3^{o}C. PMV was found to vary from -2, occurring only at the diffuser location, and 0.0

to 1 in the occupied zone. This resulted in PPD of 100% at the diffuser location but in the occupied zone it was 13.5% overall.

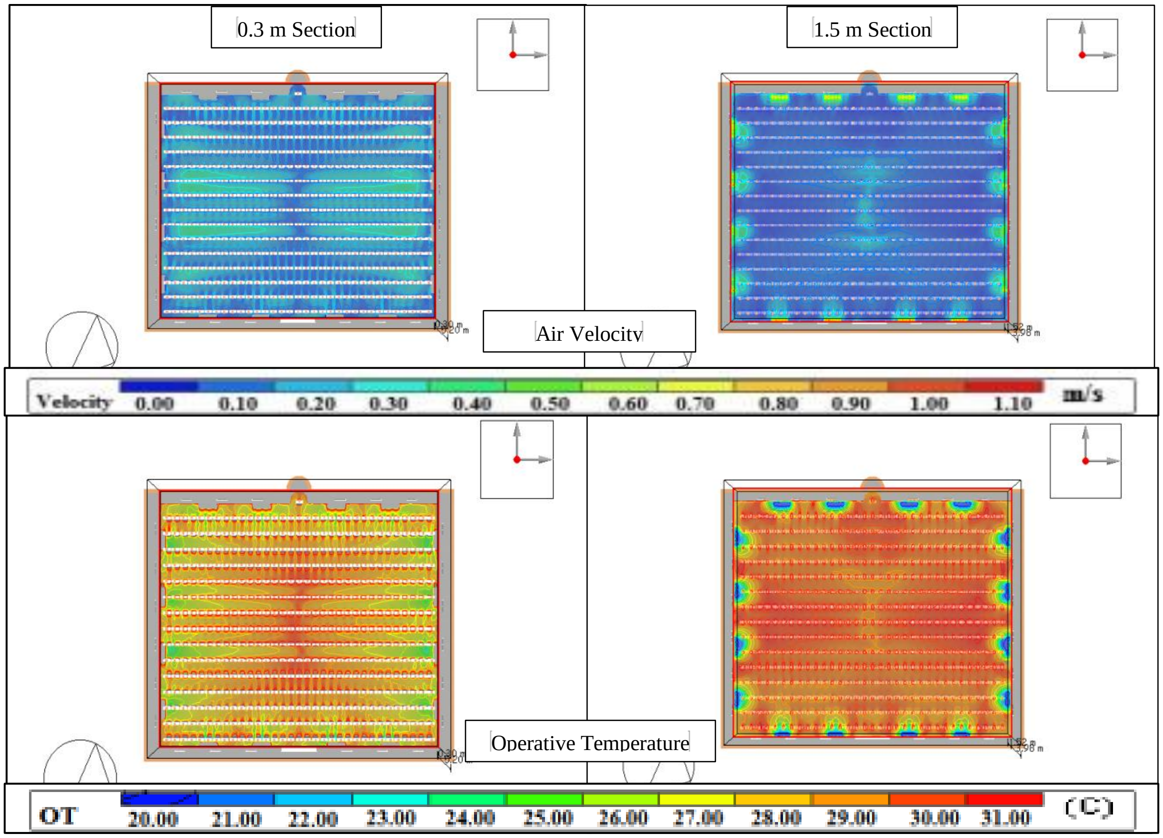


Figure 5.60: M6- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

Although the configuration attain the required 20% overall PPD to satisfy the thermal comfort criteria, there are few severe cold areas existing in the occupied zone. However, these cold spots are in the perimeter areas only. It is acceptable to say that this configuration is comfortable because in mosque usually a gap of 1 m is provided in the perimeter area for passage. However, there were few slightly warm spots seen to have exited in this model results.

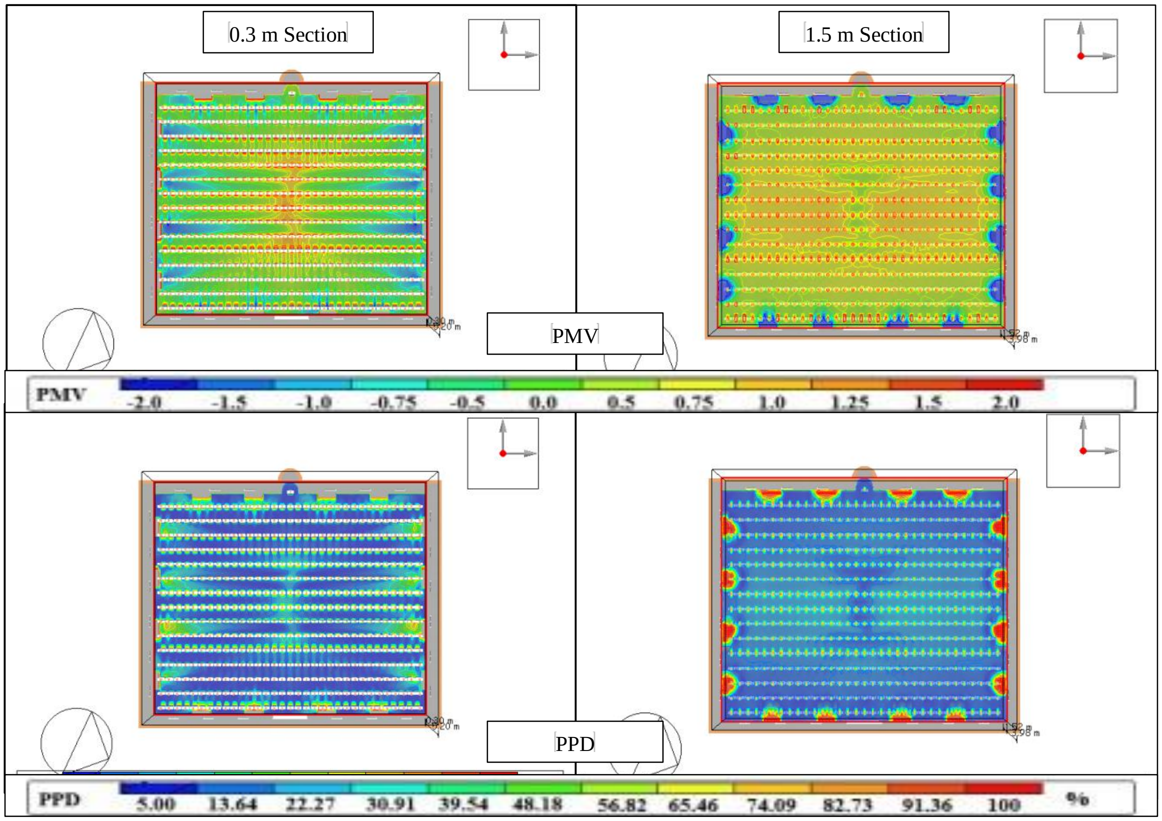


Figure 5.61: M6- PMV and PPD contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

5.5.3.2 UFAD with linear/slot supply diffusers and return on Wall (M7)

Again the setup for this model remains unchanged compared to base case model. Again the same diffuser discharge velocities, 0.8 m/s, 1.0 m/s, 1.25m/s and 1.5 m/s were used. And the results obtained for variation of diffuser discharge velocities are again very similar in each of the velocity case without any visible change. Thus results of highest velocity, 1.5 m/s are discussed here. Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.62 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.61 for a diffuser discharge velocity of 1.5m/s. There was no visible change in any parameter compared to

M6 model at the same velocity. This resulted in PPD of 100% at the diffuser location but in the occupied zone it was 13.5% overall and the slightly warm spots have disappeared in this model. This configuration achieves the required 20% or below PPD in the occupied zone. However, the clod spots near the supply diffuser location should be reduced.

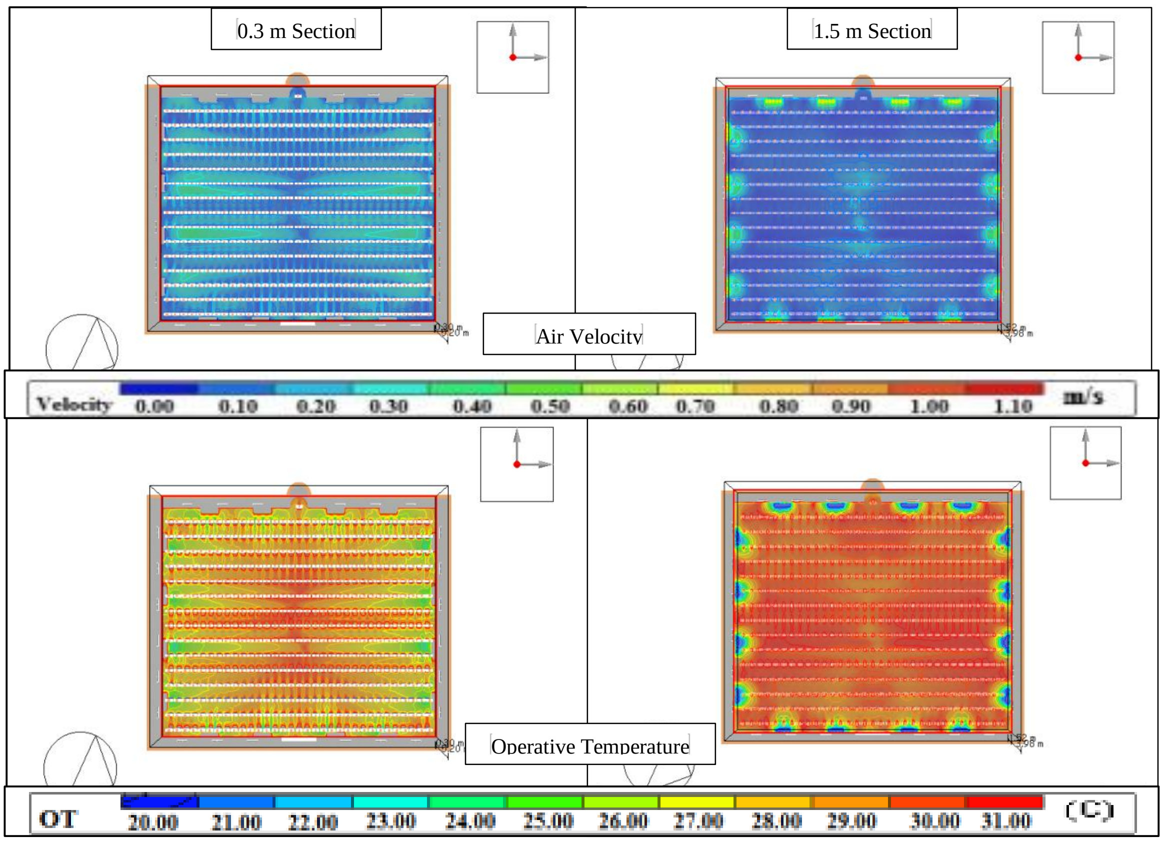


Figure 5.62: M7- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

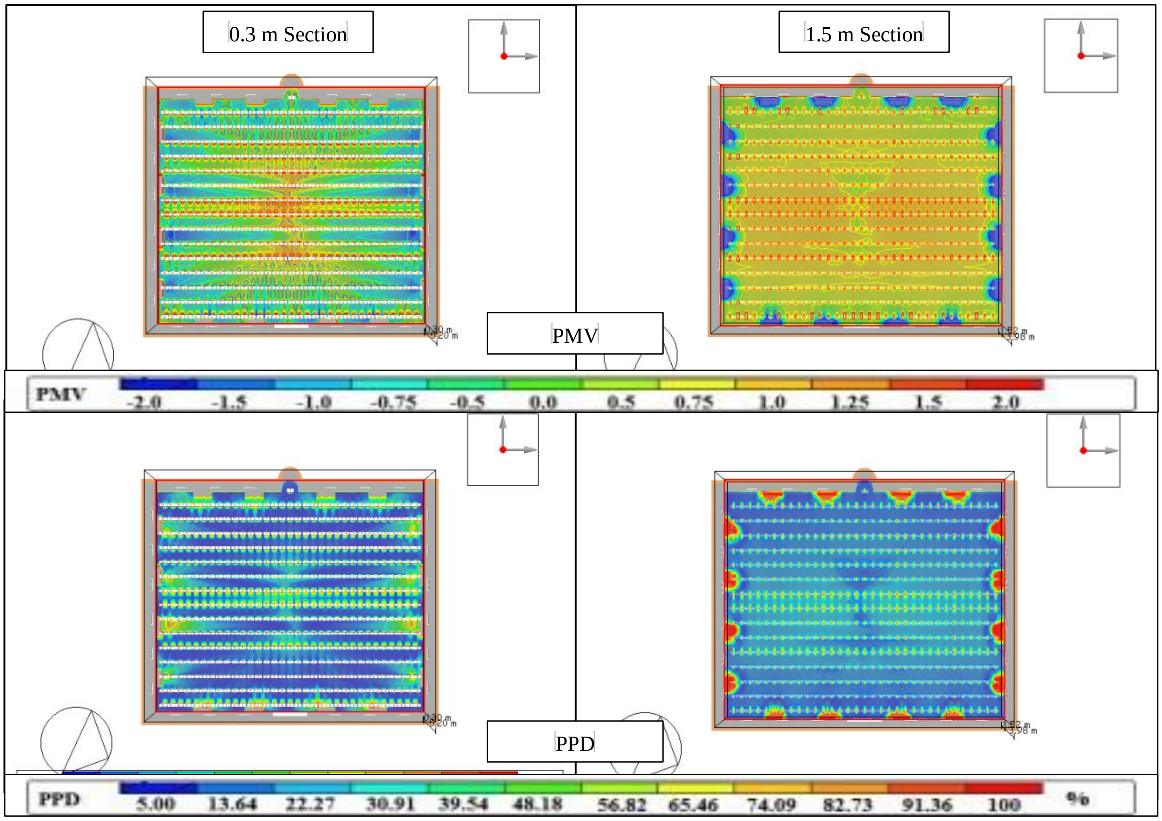


Figure 5.63: M7- PMV and PPD contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

5.5.3.3 UFAD with linear/slot supply diffusers and return on Wall (M7-1)

Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.64 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.65 for a diffuser discharge velocity of 1.5 m/s. The results of this model were very much similar to M7 model at the same diffuser discharge velocity. Air temperature in the occupied for this velocity were found to vary below 22^{o}C to 26^{o}C with 24^{o}C being predominant in the occupied zone which is within the comfort zone. Near diffuser location, it was observed to be 19^{o}C and below but in a less area compared to previous Models. In the occupied zone, air temperature stratification was within the allowable limit of 3°C. Air velocity was found to varying between 0.1m/s to 0.4m/s in the occupied zone which conforms to the comfort limit and does not allow any temperature offset. Operative Temperature also resulted towards comfort region, varying between 25°C to 29°C in the occupied zone, and a value of 20°C mainly occurring near diffusers. PMV was found to vary from -2, occurring only at the diffuser location but with less area, and 0.0 to 1 in the occupied zone. This resulted in very few spots of PPD of 100% at the diffuser location but in the occupied zone it was 13.5% overall. Having said that the problem of cold spots still exists but in reduced area. And also the practicality of this configuration is an issue as it affects construction cost and the aesthetics of the mosques.

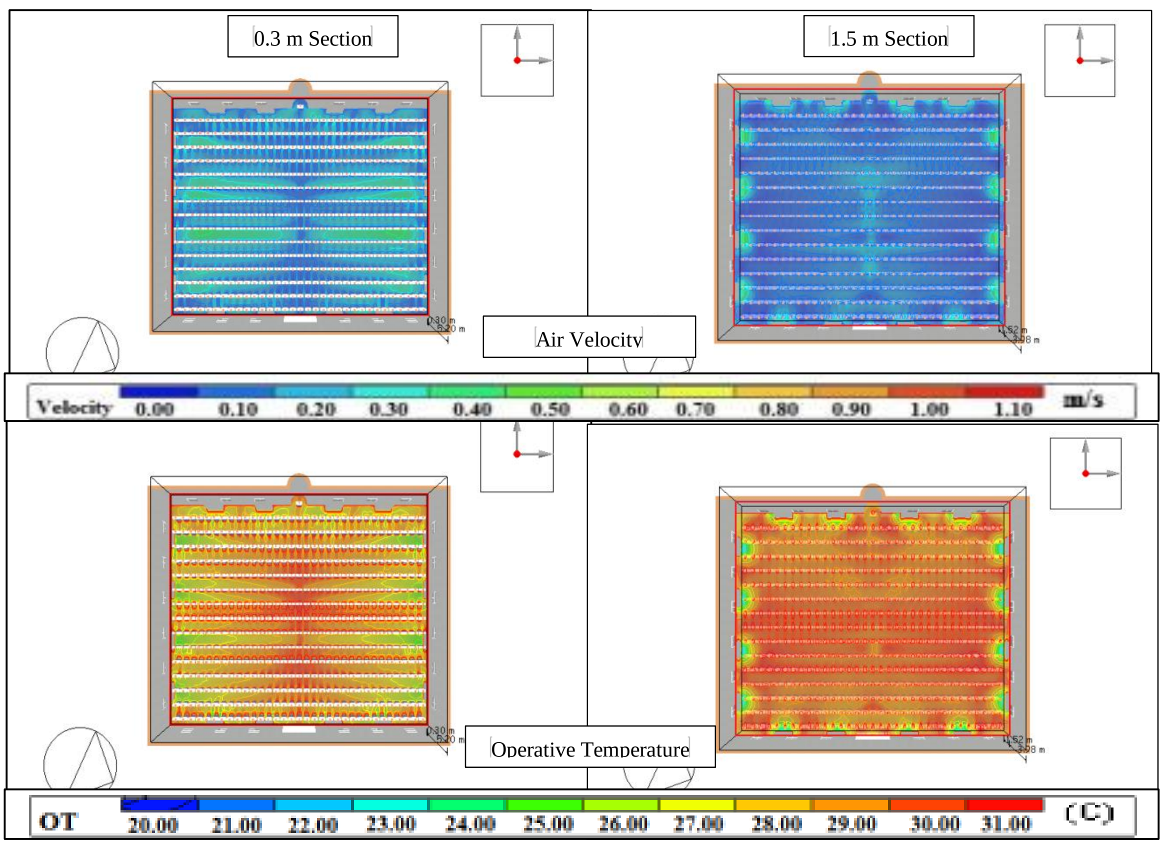


Figure 5.64: M7-1- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

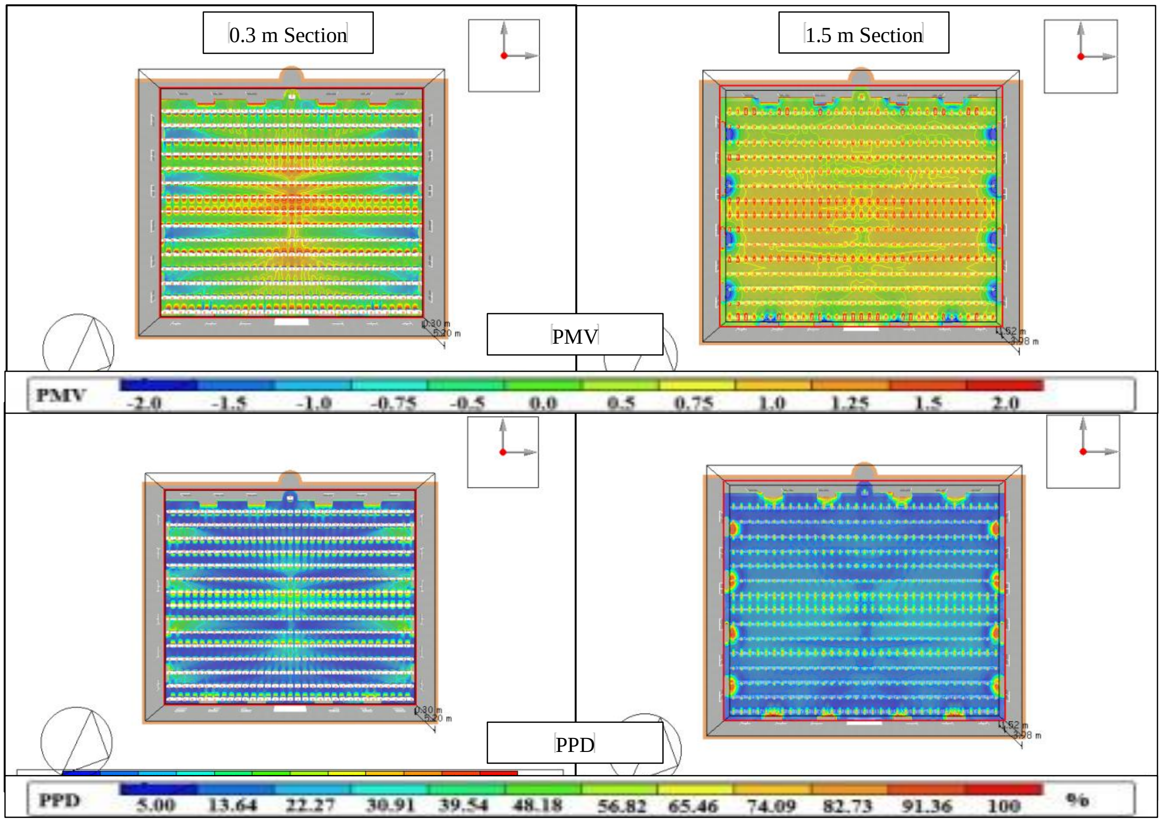


Figure 5.65: M7-1- PMV and PPD contours at sections 0.3 m and 1.5 m for 1.5 m/s velocity

5.6 Results Highlights of Intermittent HVAC Operation

The objective of this part was to test the performance of operation strategy for energy conservation and thermal comfort using energy simulation and CFD simulations. Energy simulation showed that thermal comfort performance of the operation strategy was not within the specified PMV limit of -1.0 to +1.0. But the main assumption of the energy simulation software that the air mix to a uniform temperature which resulted to 29.58^{o}C during Asr prayer time and velocity will be constant at 0.13m/s throughout the environment was not practical. In order to support this argument a detailed CFD analysis of commonly used air distribution strategies was conducted. There were 9 models created for this purpose that included 4 CBAD models, 2 TWAD models and 3 UFAD models.

Table 5.2: Summary for thermal comfort results with intermittent HVAC operation

Model No.	Parameters	Velocity Cases				Results
		Case 1	Case 2	Case 3	Case 4	
M1	Temperature (C)	24-28	24-28	25-28	26-29	Huge slightly warm spots were observed
	Velocity (m/s)	0.1-0.5	0.1-0.5	0.1-0.5	0.1-0.5	
M2	Temperature (C)	23-27	23-27	24-27	24-28	Achieved Comfort with low discharge velocity
	Velocity (m/s)	0.1-0.5	0.1-0.5	0.1-0.5	0.1-0.5	
M3	Temperature (C)	23-27	23-27	24-27	24-28	Achieved Thermal Comfort conditions
	Velocity (m/s)	0.1-0.5	0.1-0.5	0.1-0.4	0.1-0.4	
M4	Temperature (C)	22-25	22-25	23-26	23-28	Achieved Comfort with low discharge velocity
	Velocity (m/s)	0.1-0.5	0.1-0.5	0.1-0.6	0.1-0.7	
M5	Temperature (C)	22-25	22-25	23-26	23-26	Achieved Thermal Comfort conditions
	Velocity (m/s)	0.1-0.5	0.1-0.5	0.1-0.6	0.1-0.6	
M6	Temperature (C)	20-25	20-25	20-26	20-26	Achieved Thermal Comfort conditions
	Velocity (m/s)	0.1-0.2	0.1-0.2	0.1-0.4	0.1-0.4	
M7	Temperature (C)	20-25	20-25	20-26	20-26	Achieved Thermal Comfort conditions
	Velocity (m/s)	0.1-0.2	0.1-0.2	0.1-0.4	0.1-0.4	

- CBAD air distribution strategies had a reasonable performance with M2, M3 and M3-1 models. The air temperature in these three models was varying between 24°C to 27°C giving acceptable thermal comfort conditions but with few slightly warm areas. In M8 model slightly warm areas was much reduced compared to M3 but there is an issue of its practical application. The negative pressure towards return would be high compared to other locations which for this model was in occupied zone and might not be opted in reality. Model M2 also gave good results but the slightly warm spots were large compared to M3 and M3-1. The worst performance was of M1 model and to an extent it validated the EnergyPlus predictions in case of air temperature.

- TWAD strategy supplied conditioned air very close to the occupied zone, produced exceptional results and solved the problem of achieving thermal comfort during application of intermittent operation strategy. These schemes made full use of elevated air velocities that was varying between 0.2m/s and 0.5m/s which caused offset of air temperature by a value 3^{o}C, while average air temperature observed in this strategy varied between 23^{o}C to 26^{o}C. It uses the convection effect to good condition but there was short circuit problem in M4 model but M5 model performed exceptional. However, in order for M5 model to be recommended, a sensitivity analysis is required of return diffuser locations and its performance during other prayers.
- UFAD schemes also performed very well but there were cold spots near the supply diffusers which degraded its performance. As discussed earlier the supply air temperature of 12^{o}C was the reason for this col spots problem, while the literature stressed use of 15^{o}C to 18^{o}C as the supply temperature with high number of outlets possible which was out of the scope of this research work [10].

5.7 Sensitivity Analysis of Model M5

Frist the sensitivity to return diffuser location was tested using three new locations then its performance during other prayers was tested with only one supply diffuser discharge velocity i.e. 3.5 m/s.

5.7.1 Sensitivity to return diffuser location

5.7.1.1 TWAD with Wall Supply and Ceiling Return (M5-1)

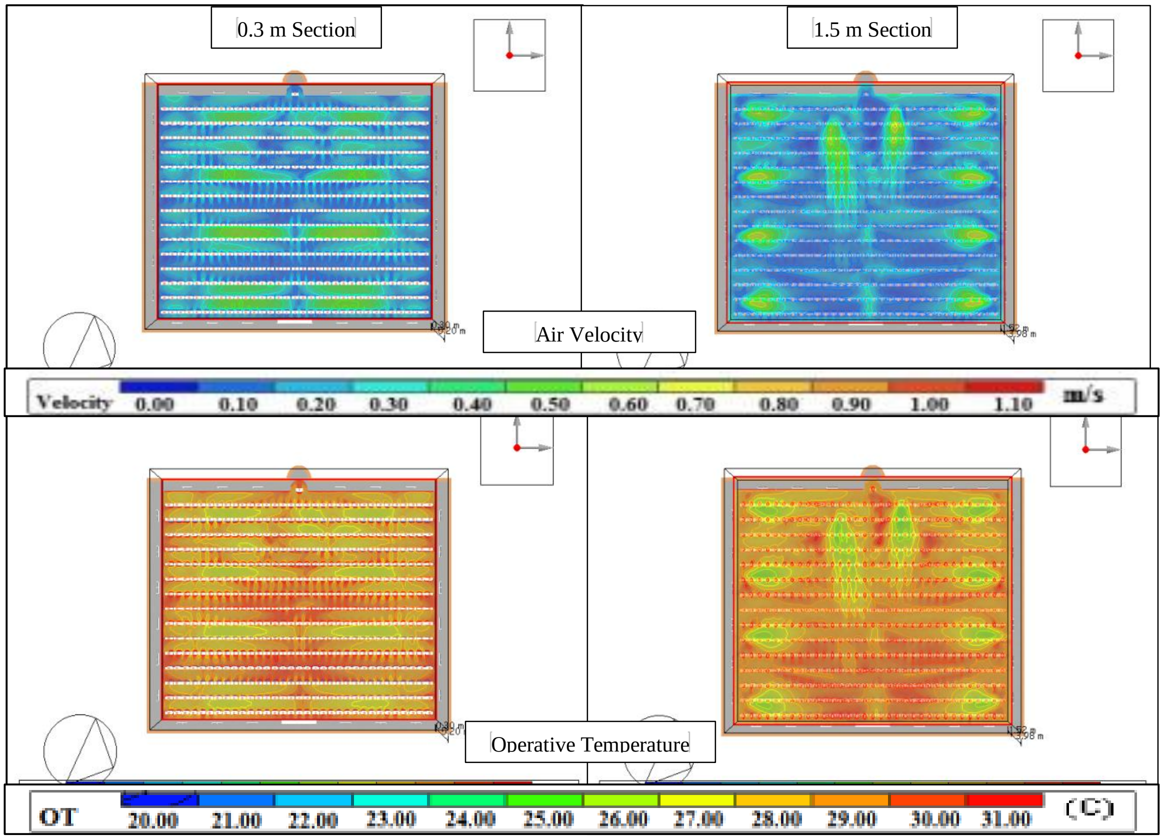


Figure 5.66: M5-1- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.66 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.67 for a diffuser discharge velocity of 3.5 m/s. In the M4 model it was observed that there was short circuiting of the conditioned air towards the return diffuser which was at same location as in this model. Similar situation has occurred in this model. There was short circuit of condition air there by creating slightly warm area near the door. The PMV was observed to vary between -0.5 to 1.0 which is a deteriorated situation. This show that the conditioned air need to sweep the whole occupied zone and then allowed to pass to the return diffuser would be a better scenario compared to this model. Thus M5-1 model reduced the efficiency of M5 by increasing slightly warm area even though the PPD was below 20% which satisfy the thermal comfort requirement.

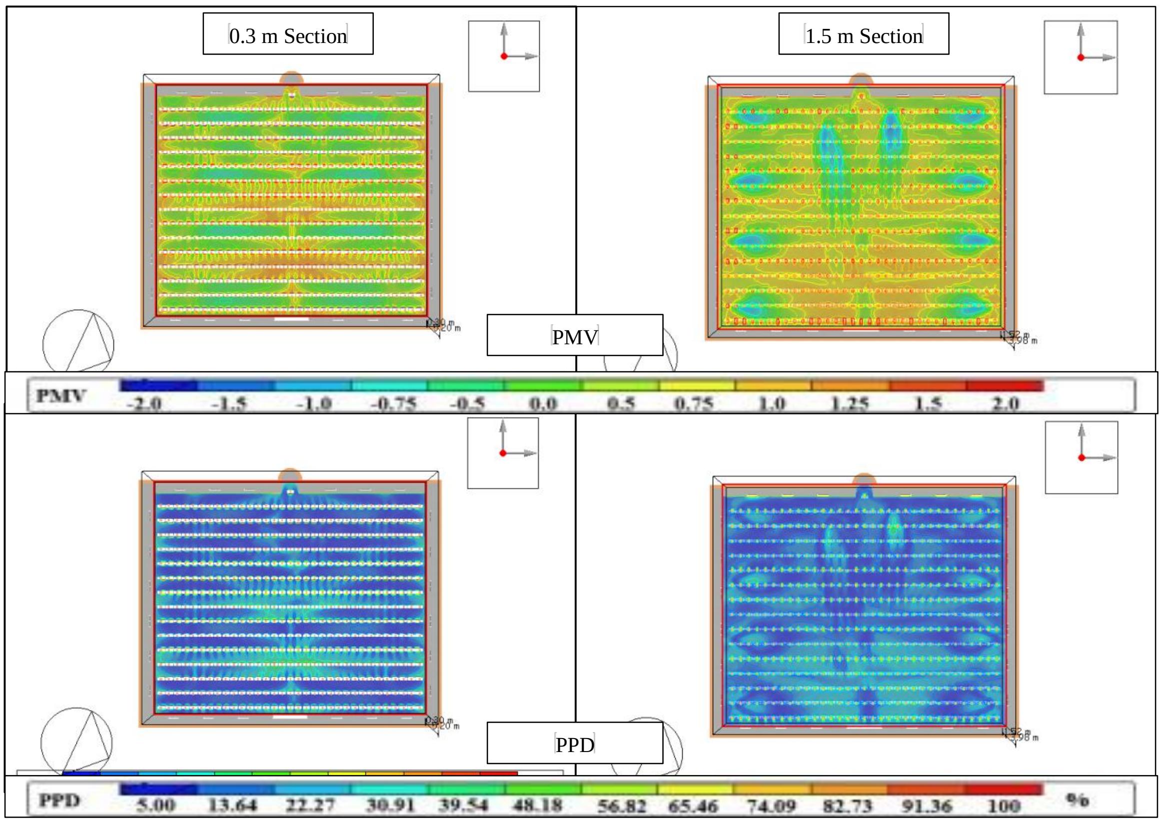


Figure 5.67: M5-1- PMV and PPD contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

5.7.1.2 TWAD with Wall Supply and Ceiling Return (M5-2)

Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.68 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.69 for a diffuser discharge velocity of 3.5 m/s. Unlike previous model there was not short circuit of condition air observed but there was slightly warm area observed below the return diffuser but not equivalent to previous case. The PMV was observed to vary between -0.5 to 0.75 which is almost the same. Even though the conditioned air swept almost all the occupied zone but buoyancy effect towards return created a warm area. Thus M5-2 model also reduced the efficiency of M5 by introducing slightly warm area even though the PPD was below 20% which satisfy the thermal comfort requirement. But this drop in efficiency is acceptable as the PPD observed in that vicinity was 22%.

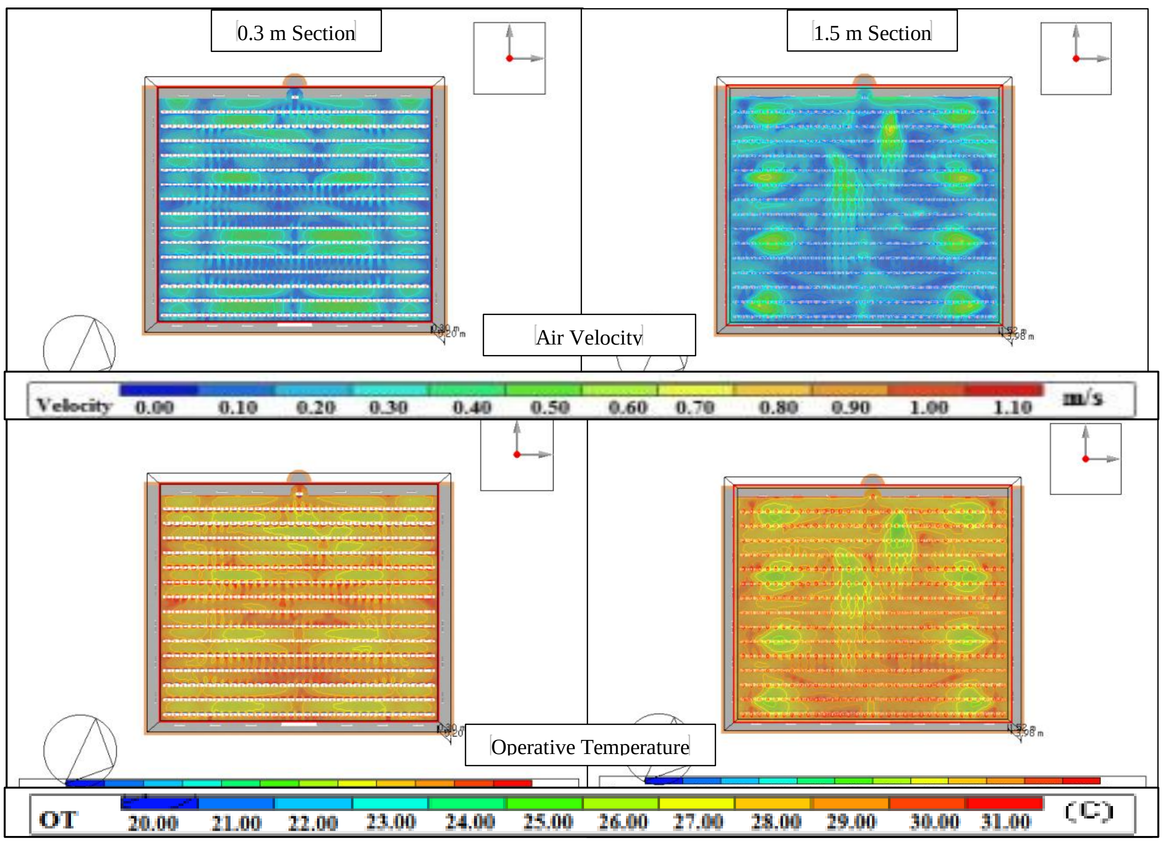


Figure 5.68: M5-2- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

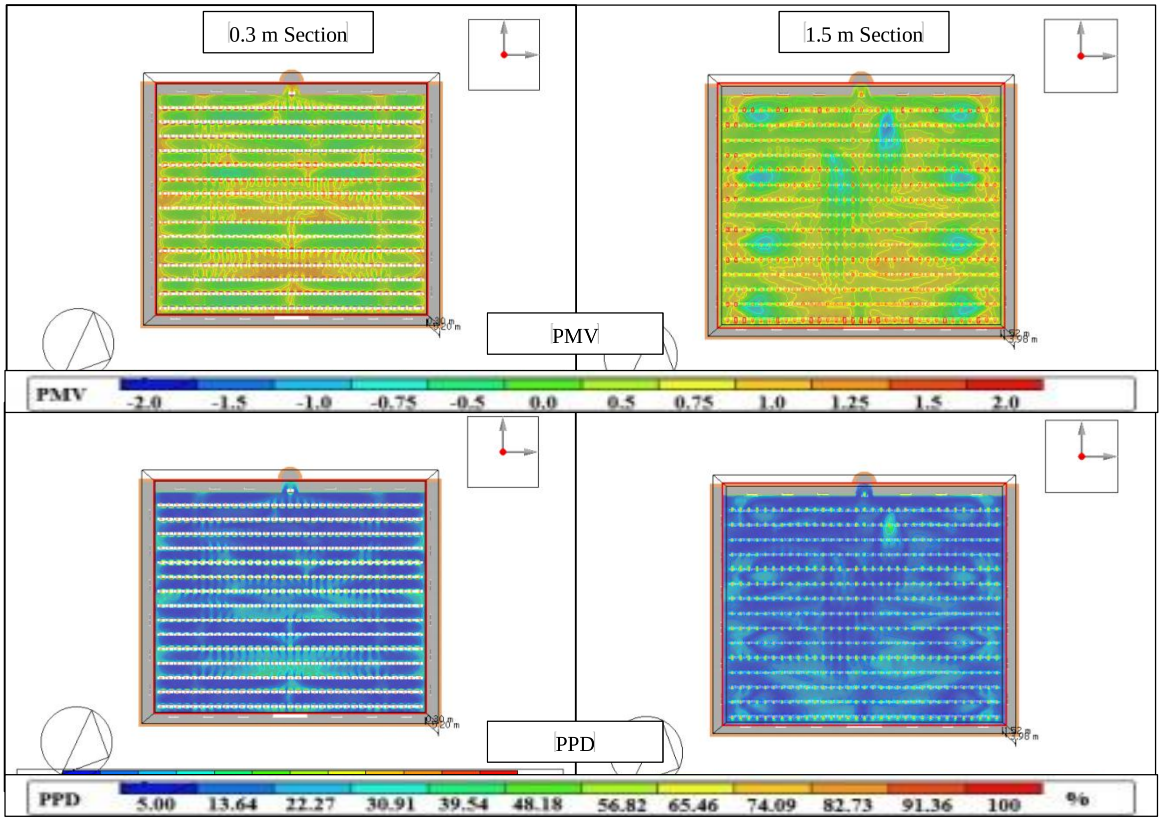


Figure 5.69: M5-2- PMV and PPD contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

5.7.1.3 TWAD with Wall Supply and Wall Return (M5-3)

Resulting Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m are shown in Figure 5.70 and resulting PMV and PPD contours at sections 0.3 m and 1.5 m are shown in Figure 5.71 for a diffuser discharge velocity of 3.5 m/s. Here thermal comfort enhanced compared to M5 model. The PMV was found to vary between -0.5 to 0.75 with very less areas of 0.75 PMV compared to M5 model. This goes to show that providing return very near to the occupied zone will enhance the thermal comfort status when using Throw the Wall supply system. However, as discussed earlier that there is an issue with its practical application.

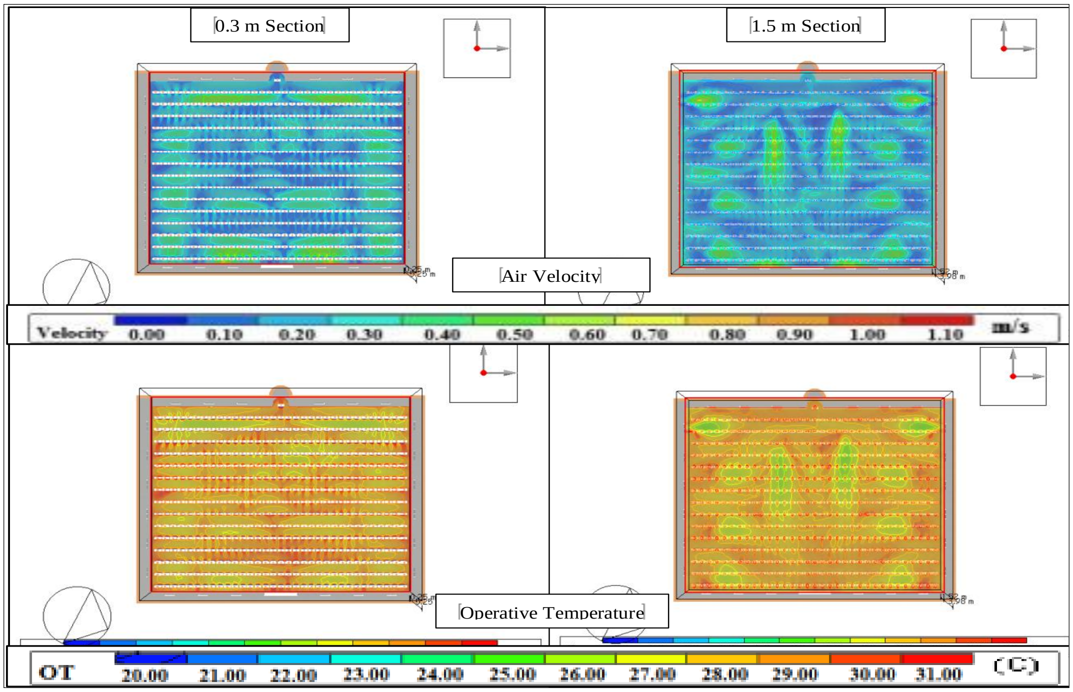


Figure 5.70: M5-3- Air velocity and Operative Temperature contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

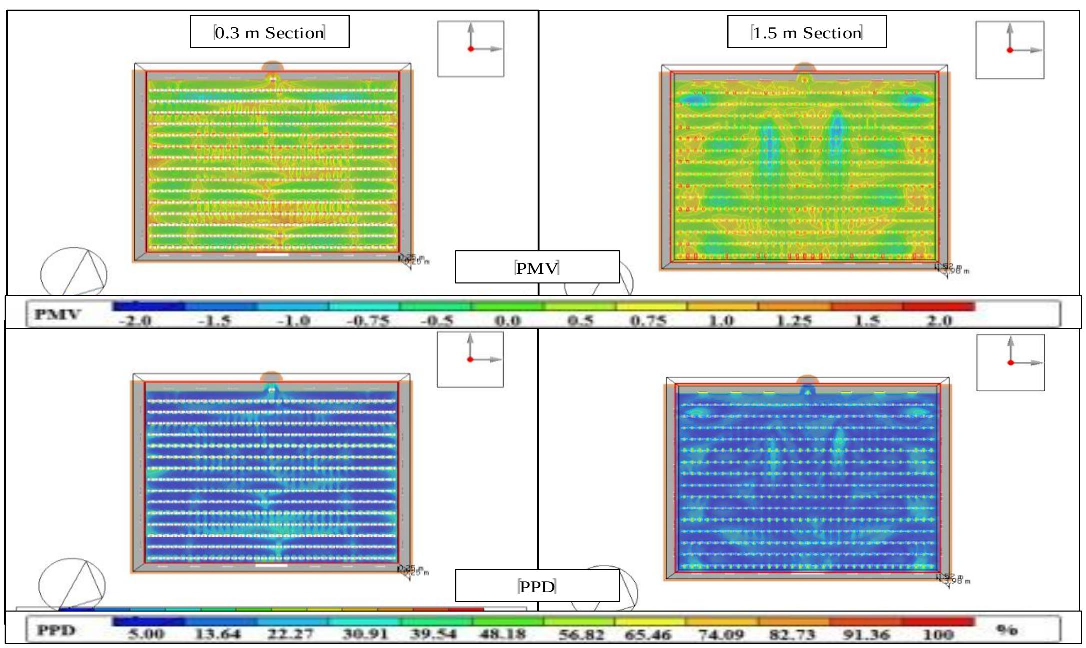


Figure 5.71: M5-3- PMV and PPD contours at sections 0.3 m and 1.5 m for 3.5 m/s velocity

5.7.2 Sensitivity to Different Prayers Time

In the operation strategy the system was starting 1 hour before every prayer except for Isha prayer as the system was kept on after Maghreb prayer till Isha ends. So the model M5 was simulated for each of these prayers by obtaining the temperature boundary conditions at 04:00 AM for Fajr, at 12:00 Noon for Dhuhr and at 07:00 PM for Maghreb. The main difference in each of these prayer timings is the sun path where in Fajr and Maghreb there is no availability of solar radiations but in Dhuhr and Asr solar is at its peak. Thus there would be difference in MRT for each of these prayers. Figure 5.72 shows the MRT for different prayer timings at 1.5 m section. It is highest During Asr at around 34°C average with west windows at 36°C, then During Dhuhr at around 34°C with all windows at around 35°C and during Maghreb it was around 33°C and during Fajr at lowest around 32°C. However, the PPD was observed to be below the required limit of 20% which shows that M5 model was working fine during all prayers (Figure 5.73).

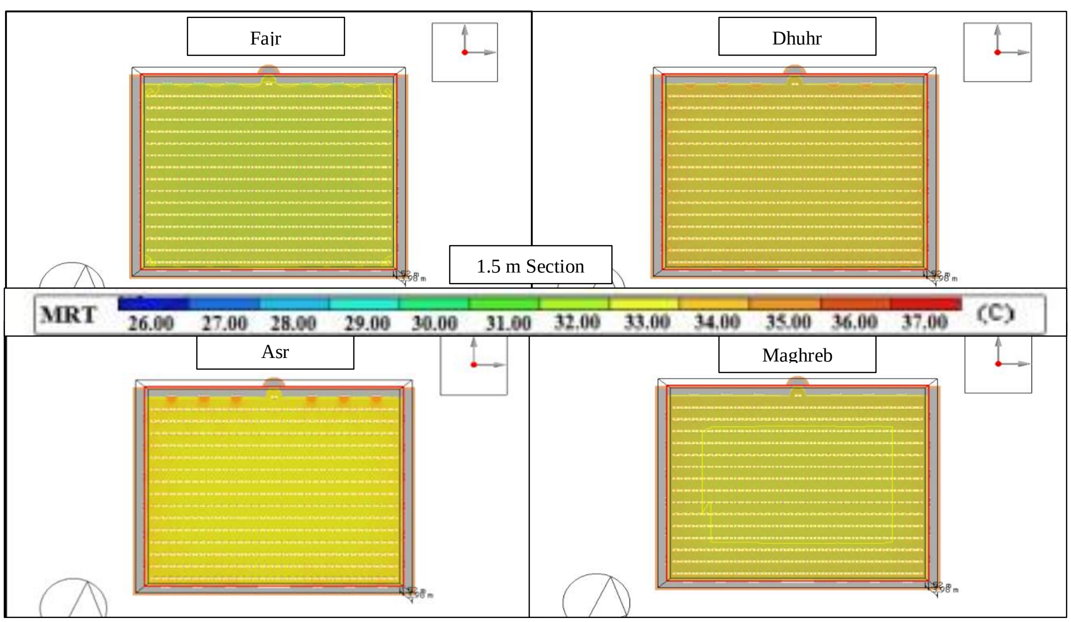


Figure 5.72: MRT at different prayers times

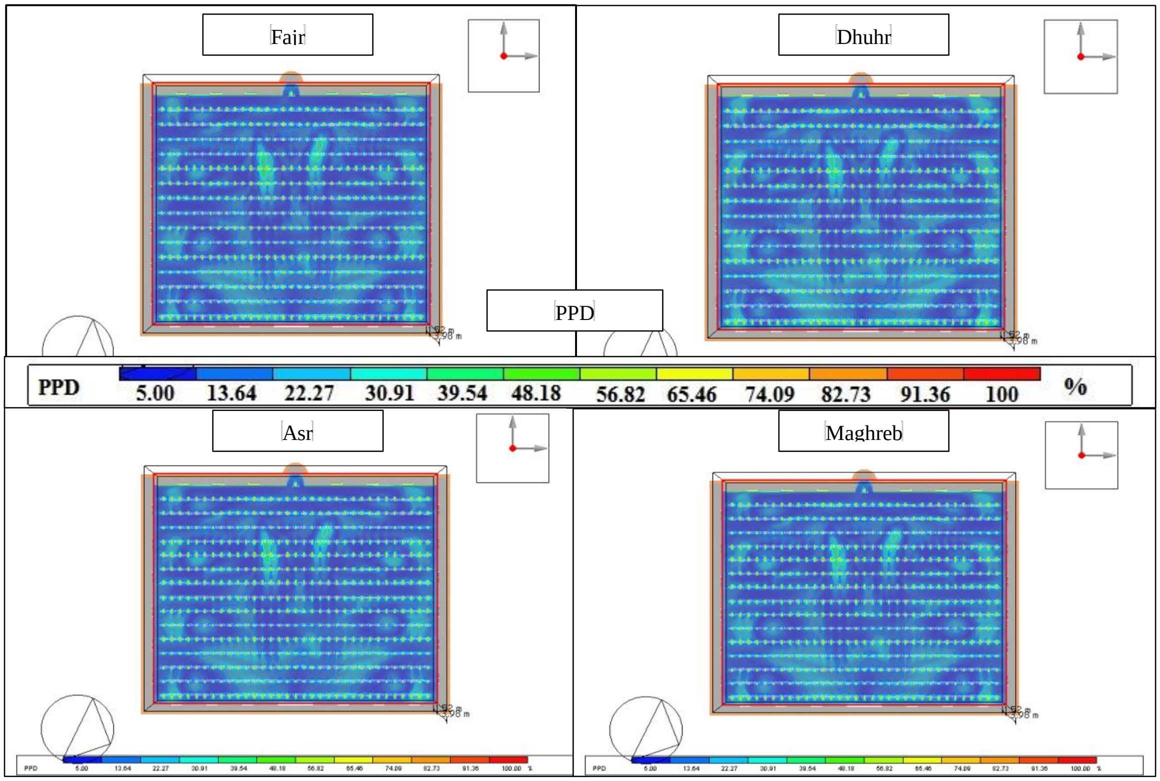


Figure 5.73: PPD during different prayers

CHAPTER 6

CONCLUSION AND RECOMMENDATIONS

6.1 Conclusion

This simulation based research work analyzed the energy and thermal comfort performance of mosque building by not only using energy simulations of EnergyPlus but also using CFD technique available in the state-of-the-art software DesignBuilder, which is first of its kind that would be applicable to Hot and Hot-Humid climates. From this study following conclusions have been derived:

6.1.1 EnergyPlus Simulation Results

- It is very difficult to achieve the set point air temperature and 0.13 m/s air velocity uniformly throughout the space. Thus EnergyPlus is not a good tool to predict Thermal Comfort.

6.1.2 Continuous Operation

- The occupied zone was found to be overcooled in most of the occasions even though the EnergyPlus simulations showed uniform thermal comfort.
- M3-(Slot/Linear ceiling supply and wall return at 3m from ground) achieved thermal comfort
- Design set point temperature required to be reassessed based on air distribution type to save energy.

6.1.3 Intermittent Operation

- Saved 30% of the total annual energy consumption by reducing the consumption from 181 kWh/m^2 to 127 kWh/m^2 consequently saving 35% of the total cooling energy consumption.
- Thermal comfort is achieved during application of intermittent operation strategy with all three air distribution schemes.
- M3-(Slot/Linear ceiling supply and wall return at 3m from ground), M5-(Through-Wall supply with 10 diffusers and wall return at 3m from ground), M6-(Under-floor supply and ceiling return) and M7-(Under-floor supply and wall return at 3m from ground) achieved thermal comfort

6.2 Recommendations

In the future design or when retrofitting the HVAC system design, following recommendations are made to ensure proper thermal comfort conditions inside the mosque buildings:

6.2.1 Continuous operation

- Mosques of all sizes with ceiling height 5 m or more that has continuous HVAC operation use Ceiling-Based Air Distribution Scheme (linear/slot type diffusers along the perimeter and wall return) to ensure thermal comfort conditions while operation.

6.2.2 Intermittent Operation

- Medium and small sized mosques utilizing intermittent operation for HVAC system should consider using Wall Based Air Distribution Scheme (supply from wall and return on wall) as air distribution system to achieve thermal comfort during all prayers.

- The diffuser discharge velocity should be selected for maximum throw with noise criteria considerations..

- For large sized mosques when utilizing intermittent operation, should use a combination of air distribution scheme like Wall distribution scheme with under-floor distribution scheme.

6.3 Future Work

This research work had few limitations in the scope, thus requires further enhancements in this field of study. The following points would summarize future work to extend the applicability of Air Distribution strategies:

1. Different air distribution schemes should have a specific design set point temperature. Thus, thermal comfort needs to be assessed using higher set point temperatures for individual air distribution schemes.
2. Air distribution schemes need to be assessed at a wide range of supply air temperatures. Especially under-floor air distribution system which has lot of potential for energy conservation by temperature stratification.
3. Other building type like auditoriums, theaters etc. need to be checked for thermal comfort.

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